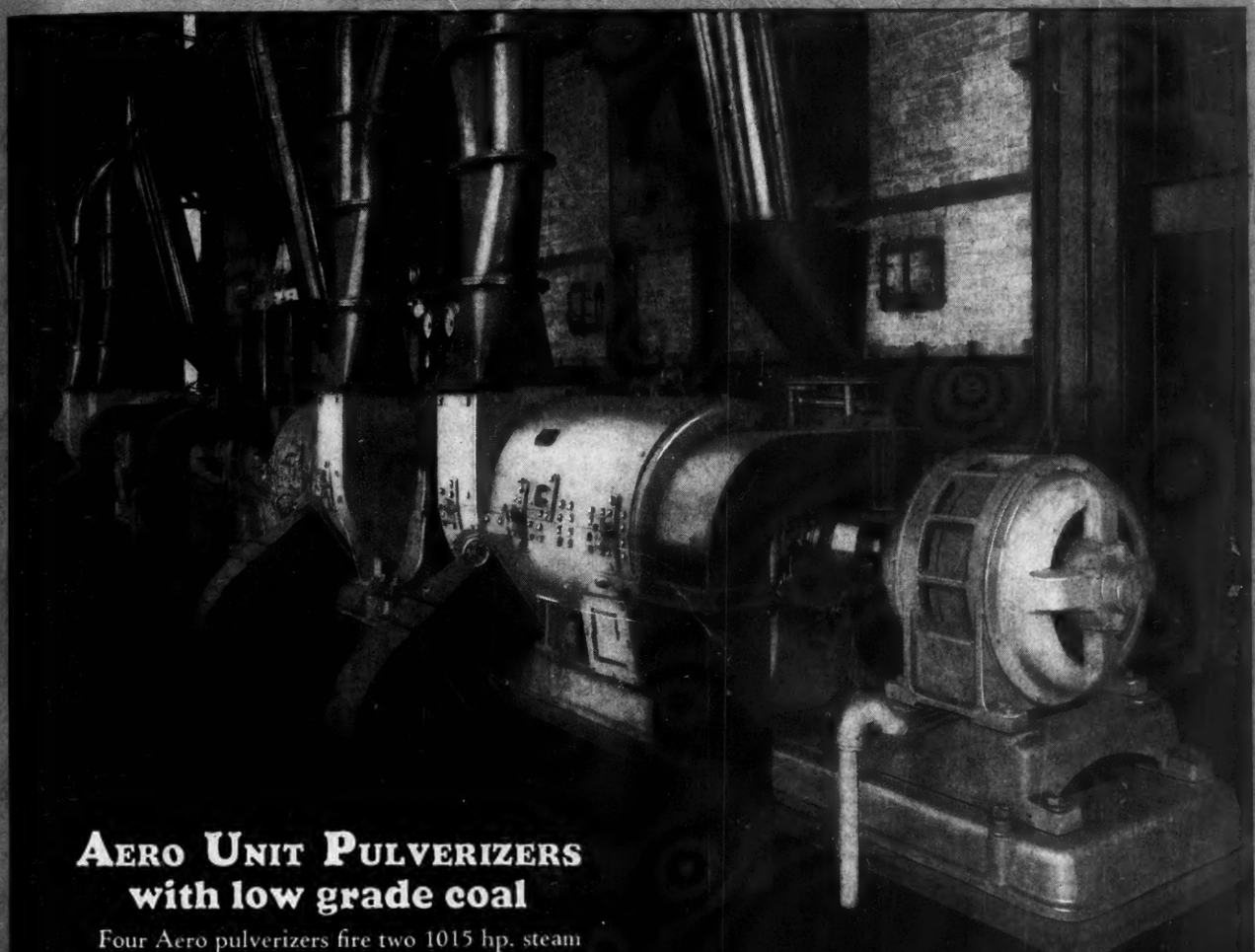
HIGH PRESSURE DEVELOPMENT NUMBER

MAY 29 1929

POWER

Power Generation, Transmission, Application and Attendant Services in all Industries

McGraw-Hill Publishing Company, Inc. MAY 28, 1929



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POWER

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Volume 69 Number 22

May 28, 1929

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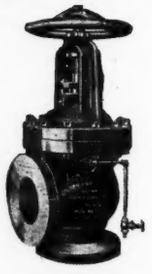

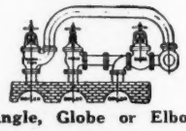

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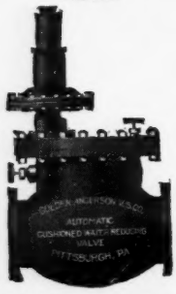

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3. No hammering, chattering or sticking.
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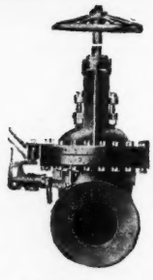

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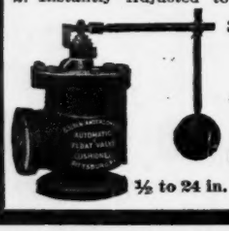

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4. No waste of water.
5. No water hammer or shock.
6. Angle or Globe patterns to 24 in.

Pat. Automatic Double Cushioned Check Valves

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2. For High or Low Pressure.
3. Thoroughly Cushioned. No Chattering, Hammering or Sticking.
4. Globe or Angle Patterns up to 24 in.
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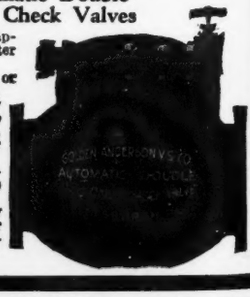

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POWER

FRED R. LOW, *Editor*

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High Steam Pressures

WITH the development of the central station, where power in the form of electricity is the only product, reduction in the cost of its generation became of prime importance.

In the design of the stations themselves and in the powerful corporations that have arisen to supply their apparatus, the best brains and talent available, backed by unlimited means and opportunity for research, experiment and demonstration, have increased in a few decades the efficiency of the processes by which power is produced from fuel more than it had been previously increased since the time of Watt.

One of the principal factors in this reduction has been the use of higher steam pressures.

At the first meeting of the American Society of Mechanical Engineers in 1880, Allan Stirling gave the average pressure then carried in practice as about 75 pounds. Fifteen years later Dr. R. H. Thurston published a diagram showing the maximum pressure then in use to be about 200 pounds.

Now pressures of from 400 to 600 pounds are common and a number of installations are successfully using the pressures of between 1000 and 2000 pounds with which Perkins and Albans experimented in the first half of the last century.

This development, if left to the industrial plant, in which power is often a minor consideration, would have been long in coming. But being here a demonstrated existing practical actuality, it holds out to the manufacturing plant the most attractive possibilities.

It costs little more to make steam at high than at low pressures. The plant that has use for steam for heating or process work can generate it at high pressure, run it through an engine or a turbine and exhaust it at the pressure required for its other services, getting its power practically as a byproduct.

This issue of *Power* is devoted to the subject of High-Pressure Steam. It treats of the boilers required to generate it, the pumps required to feed them against such pressure, the turbines which use this steam. Details are given in a table of some 140 industrial plants that have taken advantage of its possibilities.

F. R. Low

HIGH-PRESSURE POWER PLANTS WIN PERMANENT PLACE

Experience of Central Stations and Industrial Plants Demonstrates Soundness of Trend

AVERAGE PRESSURES have never receded in the whole history of steam power since the days of James Watt, who operated his condensing engines at atmospheric pressure. He called Richard Trevethick a murderer for proposing sixty-pound steam pressure and tried to enjoin him with the aid of Parliament.

Since Watt's days, pressure (considering always the average and not occasional experiments) has ascended in a series of steep steps followed by long level stretches.

Four years ago, at Edgar Station, came the spectacular jump to twelve hundred pounds. What looked to many engineers like a wild experiment, has justified itself commercially. Today, largely because of this pioneer work, three central stations in this country are operating at pressures above twelve hundred pounds, four more are under construction and many others under consideration. Many industrial plants are operating in the range from three hundred to five hundred pounds, a few from five hundred to one thousand pounds, and one is under construction for eighteen hundred pounds.

But the first cost and fuel economy do not tell the whole story. Engineers insist on asking embarrassing questions about safety, reliability and ease of operation. In answer, three operators of twelve hundred-pound plants (one industrial and two central stations) join in this issue, in an operating symposium. Significantly, each of the authors concludes that high-pressure operation has justified itself on a strictly commercial basis.

How far has the high-pressure movement progressed in terms of actual installations? Is the much-talked-of development in the field or chiefly on paper? The answers will be found in two tables in this issue, covering industrial plants and central stations.

The mere size of these lists is surprising, but they have an added significance in that they show a distinct difference between the trend for central stations and that for industrial plants. Central-station pressures show a gap between six hundred and fifty pounds and twelve hundred pounds, with no plants above fourteen hundred pounds; whereas industrial plants are sprinkled along the whole range up to eighteen hundred pounds, although concentrated largely between three hundred and five hundred pounds.

The meaning of this is that the cards are pretty well stacked for twelve hundred pounds in the case of the central station. Above five hundred pounds reheating will be necessary in any case, with present steam-temperature limitations, so why not run the pressure up to twelve hundred pounds and get some real benefit from it, particularly since twelve hundred-pound plants (condensing) cost little more than four hundred-pound plants? This is the reasonable conclusion to be drawn from the analyses of the design of two fourteen hundred-pound plants (Holland and South Amboy) published in this number.

The industrial plant is confronted by entirely different conditions. With high back pressure for process work reheating does not ordinarily enter the problem. On the other hand, the cost of a non-condensing plant increases substantially for higher pressures. Finally, the savings obtainable by higher pressures are enormously greater for the industrial plant up to the point where all power is produced as a byproduct. Beyond that point no more fuel can be saved and there is no incentive for investment in higher-pressure equipment. The result is obvious. Plants will naturally divide themselves into groups, according to the ratio of power and process demands, and each group will tend toward a certain average steam pressure, which may be anything from two hundred to two thousand pounds or more.

WHILE STEAM PRESSURES have steadily advanced, temperatures have in general remained practically the same during the last few years. Practice in this country has set about seven hundred and fifty degrees as the design limit, although uncontrolled temperatures have in many instances greatly exceeded this, usually without, but occasionally with, detriment to the equipment. Abroad, a few plants have successfully employed somewhat higher temperatures, one plant in Belgium having operated for several years at considerably over eight hundred degrees.

Announcement by Mr. Parker, in this issue, that the Detroit Edison Company is to experiment with a unit at one thousand degrees under actual service conditions will be welcome news to those who have long debated the advisability of employing higher tempera-

tures. It will be noted that the high temperature will be accompanied by a moderate pressure, around four hundred pounds, thus keeping in line with the contention that creep in metals is a function of the product of pressure and temperature.

THE DESIGN of high-pressure boilers in this country has followed closely the same general arrangement of those of moderate pressure. This has been largely due to high construction cost, which has made experimenting with new boiler designs an unattractive undertaking, and has forced the construction of boilers large enough to form an integral part of existing moderate-pressure plants. This contrasts with foreign practice, where lower costs have permitted experiments with new types such as the Benson, Loeffler and Atmos.

High cost of high-pressure boiler surface has made it desirable to reduce to a minimum the amount of this surface required for a given steam output. Thus, the high rates of evaporation secured from the extensive use of water walls helps in the solution of the problem. To decrease further the boiler surface, a greater portion of the heating surface necessary to obtain low exit gas temperature is used in the less expensive air preheater and economizer.

Circulation must be considered in the design of any boiler, but in high-pressure boilers greater attention must be given to it. In fact, improper circulation appears to have been the cause of the boiler trouble so far encountered. These problems and the design complications occasioned by the thick-walled forged-steel drums used in boilers carrying more than about seven hundred pounds pressure are enlarged upon by J. B. Crane.

The development of centrifugal feed pumps for discharge pressures of twelve hundred to sixteen hundred pounds has not proved an easy task. The designer is confronted with conflicting mechanical and hydraulic requirements, which high pressures have greatly emphasized. These limit the capacity of thirty-six hundred revolutions per minute high-pressure pumps to about nine hundred gallons per minute, and for capacities greater than this two pumps in series have been found necessary.

Exact computation of hydraulic thrusts has not been possible; and thrust bearings, in addition to hydraulic balance, are used to take care of these unpredictable forces. The influence of these and other problems on the design of high-pressure pumps is discussed in this issue by O. H. Dorer.

No radical changes in design of boiler auxiliaries such as safety valves, water columns and blowdown valves has been necessary. High excess feed-water pressures encountered in high-pressure boiler operation have, however, required the installation of excess-pressure regulators and, in some cases, the modification of feed-water regulator design.

Although it was expected that extremely high steam pressures might involve unforeseen turbine difficulties, experience thus far, aside from some early packing troubles, has indicated little that is not encountered with moderately high pressures and temperatures. Within the whole range from four hundred pounds and seven hundred and fifty degrees up, uneven distribution of temperatures has to be closely watched in order to avoid casing and spindle distortion.

As machines have become larger the blades of the low-pressure section have been subjected to extremely high mechanical stresses and vibration. Material that formerly was satisfactory is no longer suitable. Many alloys are being tried out, but as yet practice is far from settled. Certain of these alloys, while satisfactory from the viewpoint of operation, are difficult to work. Mr. Houghton's article gives an intimate picture of the situation from the viewpoint of the operator.

Uncontrolled steam temperature has been another factor with which the operator has had to contend. In one or two of the later stations an attempt has been made to control this by employing a desuperheater that will function automatically, as far as the high-pressure element is concerned, and by providing steam and gas reheaters in series for the steam to the low-pressure element.

METAL USED in the construction of boilers for pressures around twelve hundred pounds need not differ from that already in common use. Regular boiler steels have proved satisfactory for this service. Alloys that have better qualities do not lend themselves to easy fabrication and their cost is much higher. Present requirements as far as the boiler is concerned, therefore, need cause no worry.

With accessories such as superheaters, pumps, valves and fans, and in the construction of turbines, it has been found necessary to go to special and, in some cases, costly alloy metals. Problems of strength in moving parts at high temperature, coupled with the need for resistance to erosion and corrosion, render the search for satisfactory materials for these purposes a difficult job. In some cases, suitable metals have been found, while in no case is the problem so acute that progress from four hundred pounds to twelve hundred pounds is blocked. The present status of the materials of construction is covered by articles in this issue by Messrs. White, Clark, Spring, Kanter, Pigott and Straub.

From the foregoing it becomes evident why *Power* is devoting an entire issue to this subject of high pressures and high temperatures.

High Pressure Industrial Plants

ON MARCH 26 P. W. Swain, Associate Editor of *Power*, addressed the Philadelphia section of the A.S.M.E. on the subject of high steam pressures in industrial plants. In connection with the paper he presented a preliminary table of industrial plants in the United States and Canada, operating at pressures of 300 lb. and up. This served as a nucleus for building up the far more complete table reproduced herewith.

The plants listed are arranged in four groups as follows, according to boiler pressures: 1,000 lb. and up; 600 to 999 lb.; 400 to 599 lb.; 300 to 399 lb. Within each group the arrangement is alphabetical.

A few plants reported have been omitted because of inability to check the actual pressures. Information on certain other plants came too late for insertion in the table. These omissions are relatively few and it is believed that the table is a fairly complete survey of the high-pressure industrial field. Acknowledgement is hereby made of the invaluable co-operation of plant operators and manufacturers who furnished data.

Name of Plant	Boilers					Superheaters	Economizers	Air Heaters
	No.	Date Install	Make and Type Size, Sq.Ft.	Press.	Temp.			
Philip Carey Mfg. Co., Lockland, Ohio	2	Constr.	C. E., Steam Gen.	1800	200 s	Yes	Integral	Yes
Consolidated Ashcroft Hancock Co., Bridgeport, Conn.	1	1924	B. & W., cross drum	1250	800 t	Yes	No	No
Crosby Steam Gage & Valve Co., Boston, Mass.		Future		1500				
Deep Water Station, Deep Water, N. J.	6	Constr.	B. & W., cross drum, 9000	1400	725 t	B. & W.	B. & W.	Foster
Masonite Corp., Laurel, Miss.	2	1926-30	B. & W., cross drum 4000	1200	Sat	No	No	No
Ashton Valve Co., Cambridge, Mass.	1	1926	C. E., Heine, 1680	650				
Champion Coated Paper Co., Hamilton, Ohio		Future		650	750 t	Yes	No	Yes
Hancock Inspirator Co., Boston			B. & W.	650				
Solvay Process Co., Syracuse, N. Y.	2	1927	C. E., Steam Gen. 6598	804	750 t	Elesco, 3670	C. E.-4480	C. E.-23,400
Waldorf Paper Products, St. Paul	2	1929	Edge Moor, 13,460	650	155-265 s	Foster	Integral	Edge Moor
West Virginia Pulp & Paper Co., Tyrone, Va.	2	Constr.	Springfield, 8460	600				
Advance Bag & Paper Co., Hodge, La.	3	1927	C. E., W. & W., 7500	450		Foster		
Ajax Rubber Co., Racine, Wis.	1	1928	Wickes Bent Tube 8050	400	600 t	Foster	Green 6550	
American Salpa Corp., Spotwood, N. J.	2	1929	Badenhausen, 4000	402	600 t	Baden	Integral, 1250	
American Tobacco Co., Reidsville, N. C.	3	1928	B. & W., Stirling, 4690	450	604 t	B. & W.	No	B. & W.
Anglo Canadian Pulp & Paper Mills, Quebec City	3		B. & W. 12,500	400				
Anheuser Busch Co., St. Louis, Mo.	3	Constr.	Springfield, 10,760	500	600 t	Elesco		Prat-Daniel
Atmospheric Nitrogen Corp., Hopewell, Va.	3	1928	B. & W. & Erie City, 12,840	450	720 t	Elesco		
Backus Brooks Co., Westfort, Ont.	2	1928	C. E., V. X., 9990	450		Elesco		
Bogalusa Paper Co., Bogalusa, La.	1	1926	C. E., W. & W., 10,220	450		Elesco		Ljungstrom
The Buda Company, Harvey, Ill.	1	1928	C. E., W. & W., 7780	450		Yes		
Canadian International Paper Co.			B. & W.	450				
Canadian Sugar Refining Co., Montreal				425				Ljungstrom
Celanese Corporation, Cumberland, Md.	4	Constr.	B. & W. Stirling, 10,000	450				
Champion Fibre Co., Canton, N. C.	2	1928	C. E., 24,390	450	250 s	Elesco	Integral, 6000	C. E., 31600
Champion International Co., Lawrence, Mass.	2	1928	C. E., W. & W., 5520	400		Foster		
Chesapeake Corp., West Point, Va.	3	1926	C. E., W. & W.	450	100 s	Foster		Ljungstrom
Cloquet Lumber Co., Cloquet, Minn.				400				
Consolidated Water Power & Paper Co., Wis. Rapids, Wis.	4	1927	Springfield, 8150	400	450 t	Elesco	Foster, 2016	Foster, 6160
Consolidated Water Power & Paper Co., Port Arthur, Ont.				400				
Crocker, Burbank & Co., Fitchburg, Mass.			B. & W.	400				
De Laval Steam Turbine Co., Trenton, N. J.	1	1926	Badenhausen, 4210	400	550 t	Badenhausen	No	Bad., 4200
Du Pont Rayon Co., Waynesboro, Va.	3	1929	B. & W., Stirling, 8000	450	240 s	B. & W.	B. & W., 1583	B. & W. 6283
Du Pont Rayon Co., Richmond, Va.	3		B. & W., Stirling, 8000	450	240 s	B. & W.	B. & W.	B. & W.
Du Pont de Nemours & Co., Newburgh, N. Y.	1	1929	C. E., W. & W., 6330	488				
Elliott Company, Jeanette, Pa.	1	1927	B. & W.	400	750 t	B. & W.		
Garlock Packing Company, Palmyra, N. Y.	2	1928	Wickes, cross drum, 5050	400	150 s	Foster	No	No
Gates Rubber Co., Denver, Col.			6000 sq.ft.	400	250 s	Elesco	Sturt	
General Electric Company, Schenectady				400	750 t	Elesco		
B. F. Goodrich Co., Los Angeles, Cal.			B. & W.	450				
Gulf States Corp., Holt, Ala.	3	1929	C. E., V. X., 10,230	450	200 s	Elesco		
Gulf States Paper Co., Tuscaloosa, Ala.				400	200 s			
Gulf States Steel Co., Alabama City, Ala.	4	1928	B. & W., Stirling, 9068	415	250 s	Elesco	No	C. E., 8736
Howard Smith Paper Mills, Cornwall, Ont.	1	1929	B. & W., 8480	400	Sat	No	No	Sturt, 9900
Humble Oil & Ref. Co., Ingleside, Texas	3	1929	C. E., V. X., 5070	400	100 s	Foster	No	No
International Paper Co., Livermore Falls, Me.	4	1927	C. E., Ladd, 10,200	400	150 s	Elesco	No	C. E., 9000
International Paper Co., Fort Edward, N. Y.	4	1927	C. E., W. & W., 5180	460	200 s	Elesco		Sturt
Lazote, Inc., Belle, W. Va.	3	1928 and '29	C. E., W. & W., 6140	450	No	No	No	No
Mathieson Alakli Co.	2	Constr.	B. & W., Stirling, 10,040	450	600 t			Yes
Miami Copper Co., Miami, Arizona	4	Constr.	C. E., V. X., 7200	425	200 s	Elesco	No	C. E.
Minnesota & Ontario Paper Co., Int. Falls, Minn.	3	1928	C. E., V. X., 10,000	450	125 s	Elesco	No	Sturt
National Enamelling & Stamp Co., Granite City, Ill.			C. E., Steam Gen.	435	180 s	Elesco		
National Electric Products Co., Economy, Pa.	1	1926	C. E., W. & W., 7380	425	100 s	Elesco	No	No
Ninety-Six Cotton Mills, Ninety-Six, S. C.			C. E., W. & W.	400				
Nekoosa-Edwards Paper Co., Port Edwards, Wis.	2	Constr.	B. & W. Stirling, 9110	400	573 t	Foster	No	B. & W., 11,000
New Haven Pulp & Board Co., New Haven, Conn.	1	1929	Bigelow Hornsby, 7595	400	600 t	Elesco	Integral, 1639	Sturt., 6600
National Cash Register, Dayton, Ohio.								
Pittsburgh Steel Co., Monessen, Pa.	4	Constr.	C. E., Ladd, 14,250	425	200 s	Elesco	No	
Pennsylvania Salt Co., Wayndotte, Mich.	1	1928	C. E., Ladd, 12,450	450	250 s	Elesco	C. E.	C. E., 12,978
Proximity Mfg. Co., Greensboro, N. C.	2	1927	C. E., V. X., 8630	445	200 s	Elesco		C. E.
Rhinelander Paper Co., Rhindelander, Wis.	2	1929	B. & W., Stirling, 8340	415	550 t	B. & W.	Foster	No
Scott Paper Co., Chester, Pa.		Future		450	175 s			
Semet Solvay, Buffalo, N. Y.				400				
Semet Solvay, Harriett, N. Y.	2	1927	C. E., V. X., 6000	415	300 s	Elesco		
Alex. Smith & Sons, Nepperhan, N. Y.			B. & W.	400				
Paul A. Sorg Co., Middletown, Ohio.	3	1927 and '28	B. & W., Stirling, 5790	400	250 s	B. & W.	B. & W., 2969	B. & W., 7350
St. Lawrence Paper Mills, Three Rivers, Quebec.								
Standard Oil Co., Whiting, Ind.	8		B. & W., Stirling, 15,296	415		B. & W.	B. & W.	C. E., 10,584
John Strange Paper Co., Menasha, Wis.	3	1929	Casey Hedges, 6000	400	150 s	Foster	No	No
Tobacco By-Products Co., Louisville, Ky.	2	1928	Edge Moor, 2475	425	150 s	Foster	No	No
Vacuum Oil Co., Paulsboro, N. J.	3	1927	B. & W., 17,320	425				
Vancouver Kraft Co., Port Mellon, B. C.								
Viscose Co., Meadville, Pa.	3	1929	B. & W., 10,780	450	150 s	B. & W.	Future	Ljungstrom

In the United States and Canada

ABBREVIATIONS USED IN TABLE

accum.	accumulator	deconc.	deconcentrator	Man.	Manistee	St.	stoker
Asht.	Ashton	deaer.	deaerator	mix. press.	mixed-pressure	Sturt.	Sturtevant
Baden.	Badenhausen	dup.	duplex	Penn.	Pennsylvania Pump & Compressor Co.	sulph.	sulphuric
cent.	Centrifugal	Eck.	Eckenroth	Perm.	Permutit Co.	s.	superheat
Coch.	Cochrane	F. L.	Fuller Lehigh	plung.	plunger	t.	temperature
Comb. Cont.	Combustion Control	G. E.	General Electric	P. P. S. C.	Power Plant Specialty Co.	treat.	treatment
C. E.	Combustion Engineering	gen.	generator	proc.	process	V.	(Type of boiler)
Cons.	Consolidated	G. R.	Griscom Russell	soft.	softener	V X.	(Type of boiler)
Constr.	under construction	Harr.	Harrington (Riley)	S. K.	Schutte & Koerting	vac.	vacuum
Cont. blow.	continuous blowdown	h. p. soft.	hot-process softener	st. pur.	steam purifier	W. & W.	Walsh & Weidener type
Cros.	Crosby	I. R.	Ingersoll-Rand			Worth.	Worthington
		Int. Filt.	International Filter Co.			zeol. soft.	zeolite softener

Firing	Safety Valves	Blow-off Valves	Feed Pumps	Turbines						Feed Reg.
				No.	Make and Type	Kw.	Throttle P. & T.	Bleed, P.	Exh. P.	
P. C., C. E. Oil	Consolidated Consolidated Crosby			2	Trip. Exp. Engines	5000 hp.	1400 lb., 800 t		60	
P. C., C. E. and F. L. Wood	Consolidated Consolidated		Cameron, Cent Dean-Hill, Dup.	2 No	G. E.	50,000	1200 lb., 725 t		See note	Copes
Oil, Todd	Ashton	Yarway	Worthington, Dup.							
	Consolidated					8500		Bleed		
P. C., C. E. P. C., Strong Scott	Consolidated Consolidated	Yarway Yarway	Cameron, Cent.	2 1	West West	5000 3000	725 lb., 725 t 600 lb., 250 t		165 60	Copes
Gas St., Taylor St., Taylor	Cros. and Cons. Ashton	Yarway Eckenroth	Allis-Chal., Cent. De Laval, Cent.	1	West	1500	380 lb., 600 t	170	85	Copes
P. C., Fuller Bonnot St., Taylor	Consolidated Cons. and Cros.	Yarway Yarway	Allis-Chal., Cent.	1	West West	1500 7500	400 lb., 200 s 375 lb., 100 s	90 110	27-in. vac. 25	Copes
St., C. E. P. C. Strong Scott	Crosby Cros. and Cons.	Yarway Yarw. & Coch.	Dean-Hill, Cent. De Laval, Cent Allis-Chal., Cent	2	Allis-Chal. West	3750 kva. 4000	400 lb., 500 t 400 lb., 650 t	125	5 21	Stets Stets
Gas & P. C., C. E. P. C., Fuller Bonnot		Yarway Yarway								Copes
	Cons. and Cros.									
St., Taylor	Crosby Consolidated	Yarway Yarway								Copes
P. C., C. E. St., Taylor	Asht. and Cros.	Yarway Yarway	Cameron, Cent.	1		4000	400 lb., 700 t	175	25	Copes
St., West.	Consolidated	Yarway Yarway	Cameron, Cent.	1	West.	1500	420 lb, 100 s		130	Copes Copes
		Yarway	Cameron, Cent.	1 1	G. E.	6000 2500	400 lb. 350 lb., 686 t 385 lb.	125–30 110–25	28.9-in. vac. 35	Copes
	Consolidated	Yarway								
St., Taylor	Ashton		DeLaval, Cent.	4	DeLaval		400 lb.	4	29-in. vac.	
P. C., C. E.	Consolidated	Yarway	Cameron, Cent.	2	G. E.	3000	400 lb., 700 t	20–75	Cond.	Copes
P. C., C. E.	Consolidated	Yarway		2	Allis-Chal.	5000	400 lb., 681 t	30	28-in. vac.	Copes
P. C., Riley Oil			Dean Bros.	1	Elliott mix. pres.	750	150 s	2	29-in. vac.	
St., West. U. F.		Yarway	Cent. and Plung.	1	West.	1000	385 lb.	60		Copes
	Crosby		Allis-Chal., Cent.			1500				Copes
	Cons. and Cros.	Yarway								
	Consolidated	Yarway								Copes
P. C. Riley	Ashton	Yarway		1	Ferry (paper mach.)	520 hp.	400 lb., 200 s		35	Copes
P. C. and Gas, C. E.	Consolidated		DeLaval, Cent.	6	See note	See note	385 lb., 700 t	See note	1 abs.	
P. C., Fuller Bonnot	Cros and Cons.	Yarway	Cameron, Cent.	No						
Gas, Peabody			Goulds, Cent.	No						
P. C., C. E.	Cons. and Cros.		DeLaval, Cent	1	G. E.	3000	400 lb., 608 t		125	Copes
St., C. E., Coxe	Consolidated		DeLaval, Cent.							Copes
P. C., Aero	Cons. and Cros.	Yarway	Allis-Chal., Cent.	2	Ferry (pump drive)	1600 hp.	450 lb.			Copes
P. C., Fuller Bonnot	Consolidated									Stets
Oil, Peabody		Yarway	DeLaval, Cent.	1	West.	12500	400 lb., 650 t	85, 25, 5	28-in. vac.	
P. C., Strong Scott	Asht. and Cons.	Eckenroth	Allis-Chal.	2	G. E.	8000	450 lb., 700 t	135	25	
	Consolidated		DeLaval, Cent.	1	West	1000	285 lb., 200 s		155	Copes
St., Taylor		Yarway	Worthington, Dup.	1	Terry	937	400 lb., 547 t	125	55	Copes
		Yarway		1	Allis-Chal.	2000	245 lb., 554 t		27½-in. vac.	
P. C. Raymond			(Not purch.)		(Not purch.)					
		Yarway	Pennsylvania	1	West.	2000	380 lb., 150 s	80–100	20–30	
				1	West.	4000	400 lb., 200 t	3	28.5-in. vac.	
P. C. and Gas, C. E.	Cons. and Cros.	Eckenroth	DeLaval, Cent.	1	G. E.	5000	375 lb., 572 t		150	
P. C., C. E.	Ashton	Yarway	Allis-Chal., Cent.							Copes
P. C., C. E.	Asht. and Cons.	Yarway	DeLaval, Cent.	3	G. E.	8500	400 lb.	150–20	28.5-in. vac.	
St., Taylor	Consolidated	Eckenroth	Worthington, Cent.	1	West..	4000	385 lb., 550 t	125–20	28-in. vac.	
	Consolidated			1		500	400 lb., 700 t		160	
	Consolidated			1	West.	1500	400 lb., 700 t		160	
	Consolidated	Yarway								Copes
P. C., Fuller Lehigh	Consolidated	Yarway	Dean-Hill, Cent.	2	G. E.	5000	375 lb., 720 t	50–20	27.9-in. vac.	Copes
				1	West.	1500	400 lb., 150 s		120	
P. C., C. E.	Cons. and Cros.	Yarway	Allis-Chal., Cent.	3	Allis-Chal.	5000	375 lb., 623 t		125	
St., West.		Yarway	Allis-Chal., Cent.	1	Allis-Chal.	2500	385 lb., 595 t	125	15	
St., Riley		Yarway	Warren	1	G. E.	300	400 lb.	105		Copes
St., U. F.	Cons. and Cros.	Yarway	DeLaval, Cent.	1		3000	400 lb., 536 t		190	
				1	West.	1500	385 lb., 100 s	125	28-in. vac.	
P. C., C. E.	Consolidated	Eckenroth		2	G. E.	5000	450 lb., 610 t	150–15	28.5-in. vac.	Copes

Name of Plant	Boilers					Superheaters	Economizers	Air Heaters
	No.	Date Install	Make and Type Size, Sq.Ft.	Press.	Temp.			
Wausau Paper Mill Co., Brokaw, Wis.	2	1927	Springfield, 5120	450	.....	Foster	No	No
West Vaco Chlorine Prod. Co., Charleston, W. Va.	1	1927	B. & W., Stirling, 10,000	400	660 t	B. & W.	No	No
West Virginia Pulp & Paper Co., Covington, W. Va.	2	1926	Edge Moor, 8250	425	190 s	Foster	Foster, 1728	Ljungstrom
	2	1928	Edge Moor, 8250	425	190 s	Foster	Foster, 1728	Ljungstrom
C. H. Wheeler Mfg. Co., Philadelphia	1	1927	Edge Moor, 2672	425	Sat	.....	No	No
Youngstown Sheet & Tube Co., Youngstown, Ohio.	6	1929	B. & W., Stirling, 20,283	400	247 s	.....	.....	C. E., 20,160
Aberfoyle Mfg. Co., Chester, Pa.	2	1927	Badenhausen, 7500	300	10 s	No	Integral, 2700	Bad., 10,000
American Enka Corp., Asheville, N. C.	4	Constr.	.....	350	.....	.....	.....	.....
Bergstrom Paper Co., Neenah, Wis.	3	1929	Erie City 3-drum, 5650	350	160 s	Elesco	Integral	No
Branch River Wool Combing Co., N. Smithfield, R. I.	2	1924	B. & W.	300	.....	No	No	No
Bryant Paper Co., Kalamazoo, Mich.	3	1923	B. & W., Stirling, 7780	300	200 s	B. & W.	Greene	No
Canada Paper Board, Ltd., Toronto, Ont.	2	1928	B. & S., Stirling	300	150 s	B. & W.	No	Ljungstrom
Canada Sugar Ref. Co., Montreal	2	1927	Burroughs, 7656	300	70 s	Elesco	.....	Ljungstrom
Canadian National Railways, Pt. St. Charles, Quebec.	.....	.....	.....	350	.....	.....	.....	.....
Canadian Salt Co., Sandwich, Ont.	3	1926 and '29	Erie, 7000; B. & W. 10,000	350	710 t	Foster	No	
Philip Carey Magnestia Co., Plymouth Meeting, Pa.	2	1928	Badenhausen, 7500	354	450 t	No	Integral, 2700	Bad., 10,000
Central Alloy Steel Corp., Massilon, Ohio.	6	1926	B. & W.	300	600 t	B. & W.	Integral	
Colorado Fuel & Iron Co., Minnequa, Col.	5	1926	C. E., C. H., 12,000	315	200 s	Elesco	.....	.....
Crane Company, Chicago	.....	.....	.....	350	.....	Elesco	.....	.....
Chrysler Corp., New Castle, Ind.	.....	.....	C. E.	300	.....	.....	.....	.....
Dominion Sugar Co., Wallaceburg, Ont.	3	1925	B. & W., 6700	350	.....	.....	.....	.....
Dow Chemical Co., Midland, Mich.	1	1924	Wickes cross drum, 8200	385	625 t	Foster	.....	Dow, 70,000
	2	1924	Wickes cross drum, 8200	385	625 t	Foster	.....	Dow, 70,000
	3	1927	Wickes cross drum 15,000	385	625 t	Foster	.....	Dow, 175,000
	4	1927	Wickesbent tube, 15,000	385	625 t	Foster	.....	Dow, 175,000
	5	1929	Wickesbent tube, 15000	385	625 t	Foster	.....	Ljungstrom
	6	1929	B. & W., Stirling, 18000	385	625 t	B. & W.	.....	B. & W., 29,158
Eagle Paper Co., Joliet, Ill.	2	1924	Paige cross drum	300	150 s	Foster	.....	.....
Endura Mfg. Co., Philadelphia.	1	.....	Badenhausen	354	450 t	.....	.....	.....
Fort Frances Pulp & Paper Co., Fort Frances, Ont.	.....	.....	B. & W.	385	.....	.....	.....	Ljungstrom
General Electric Co., Philadelphia, Pa.	1	1929	B. & W., 5440	325	.....	.....	.....	.....
General Electric Co., West Lynn, Mass.	2	1921	B. & W.	350	100 s	Yes	.....	.....
Glenn Alden Coal Co., Nanticoke, Pa.	6	1925–'28	B. & W., Stirling	300	535 t	B. & W.	B. & W.	.....
Graham Paige Company, Evansville, Ind.	2	1928	C. E., Heine, 7560	300	.....	No	No	.....
Grays Harbor Pulp & Paper Co., Hoquiam, Wash.	5	1928	PSMD, Stirling, 8400	375	150 s	Elesco	.....	.....
Halltown Paper Board Co., Halltown, W. Va.	2	.....	C. E., W. & W., 4000	300	150 s	Elesco	No	C. E.
Hanes Knitting Co., Winston-Salem, N. C.	2	1928	C. E., Heine, 5000	300	200 s	Elesco	No	No
Jon. Heald Paper Co., Lynchburg, Va.	3	Constr.	B. & W., Stirling, 10,440	325	100 s	B. & W.	No	B. & W.
Industrial Rayon Corp., Covington, Va.	3	1929	C. E., W. & W., 6570	300	200 s	Elesco	No	Greene
Inland Steel Co., Indiana Harbor, Ind.	6	1925 and '29	C. E.	300	.....	.....	.....	.....
Int. Comb. Tar & Chem. Co., Newark, N. J.	2	1928	C. E., V., 6160	300	.....	.....	Integral	.....
McGill University, Montreal.	.....	.....	.....	385	.....	.....	.....	.....
Mead & Johnson, Evansville, Ind.	.....	1927	B. & W., Stirling	300	Sat	No	No	No
Motor Bodies, Inc., Evansville, Ind.	2	1928	C. E., V. X., 7540	325	.....	.....	.....	.....
Kenora Paper Mills, Kenora, Ont.	4	1926	B & W.	385	150 s	B. & W.	No	Ljungstrom
Nashua River Paper Co., East Pepperell, Mass.	2	1927	B. & W., Stirling	300	.....	.....	.....	.....
National Paper Products Co., Pt. Townsend, Wash.	.....	.....	B. & W.	375	150 s	Eles & Fost.	.....	.....
National Sugar Refining Co., Long Island City, N. Y.	5	1929	B. & W., 12,132	325	75 s	B. & W.	.....	B. & W.
Packard Motor Company, Detroit, Mich.	6	1924–'29	Connelly & Wickes (1)	325	150 s	Foster	No	Ljungstrom
Perry Knitting Co., Perry, N. Y.	2	1924	B. & W., 4544	300	100 s	B. & W.	No	No
Raritan Copper Works, Perth Amboy, N. J.	3	1926	B. & W. Stirling	385	200 s	B. & W.	Intergal	
Restigouche Paper Co., Athol, N. B.	2	1929	Badenhausen, 8200	350	200 s	Baden.	Baden	Baden
R. J. Reynolds, Winston-Salem, N. C.	4	1926 and '28	Edge Moore, 6646	385	200 s	Foster	No	No
Ross Pumping Station, Pittsburh, Pa.	.....	.....	.....	315	200 s	.....	.....	.....
Riverside Fiber & Paper Co., Appleton, Wis.	2	1927	Wickes, cross drum, 4080	300	.....	Foster	.....	.....
Sayles Biltmore Bleacheries, Biltmore, N. C.	2	1926	B. & W., 4770	325	550 t	B. & W.	.....	.....
Schutte & Koerting Co., Cornwells Heights, Pa.	2	1912 and '26	B. & W., 1500	350	Sat	No	No	No
Sharples Corp., Belle, W. Va.	.....	.....	B. & W.	350	.....	.....	.....	.....
Standard Oil Co., Bayway, N. J.	6	1926	Bigelow, Horns. & Springfield	380	700 t	Elesco	Bigelow Hornsby	No
Strathcona Paper Co., Strathcona, Ont.	.....	.....	B. & W.	385	.....	.....	.....	.....
Sunbury Converting Works, Sunbury, Pa.	1	1928	Badenhausen, 5400	350	10 s	No	Integral	Baden., 8000
Susquehanna Silk Mills, Belvidere, N. J.	2	1929	Badenhausen, 8750	350	10 s	.....	Baden., 3900	Baden., 12,000
Tennessee Coal & Iron Co., Fairfield, Ala.	5	1928	B. & W., Stirling, 14,607	375	200 s	B. & W.	B. & W.	Yes
Thunder Bay Paper Co., Port Arthur, Ont.	2	1926	B. & W. cross drum 10, 140	385	660 t	Foster	B. & W., 2297	B. & W., 8490
Titanium Pigment Co., St. Louis, Mo.	3	1928	C. E., W. & W., 4000	300	150 s	Elesco	.....	Ljungstrom
University of Chicago.	3	1929	C. E., Heine	300	.....	Foster	.....	C. E.
U. S. Naval Academy, Annapolis, Md.	1	1921	Navy Type P	300	200 s	Elesco	.....	.....
U. S. Gypsum Co., Oakfield, N. Y.	3	1925	Erie City 5020	325	150 s	.....	Sturt.	.....
Utica Gas & Electric Co. (Gas Plant) Utica, N. Y.	2	1926	Springfield, 4480	300	100 s	.....	No	No
Virginia Ry. Co., Narrows, Va.	5	1925	Springfield cross drum	325	150 s	Elesco	.....	.....
Warren Mfg. Co., Milford, N. J.	2	1928	Edge Moor 5120	385	100 s	Foster	Foster	No
Western Electric Co., Brill, N. J.	.....	.....	B. & W.	300	.....	.....	.....	.....
Weyerhauser Timber Co., Klamath Falls, Oregon.	4	1929	Erie City, 7960	325	.....	Foster	.....	.....
Worthington Pump Co., Harrison, N. J.	.....	.....	B. & W.	350	.....	.....	.....	.....

Notes Covering Certain Equipment Not Tabulated

Plants 1,000 lb. and Up

Deepwater — Central station. A 1,200-lb. turb. (not listed) exhausts to proc.

Masonite Corp.—1,000-lb. used in proc. proc.

Plants 600 lb. to 999 lb.

Solvay Proc. Co.—Hoppe deaer.

Waldorf Paper Prod. Co.—Coch. st. pur. Ruths accum.

Plants 400 lb. to 599 lb.

Ajax Rubber Co.—Hagan Comb. cont. Henzey deconc. Exhausts to proc.

Am. Tob. Co.—Perm. zeol. soft. and cont. blow.

Anglo Canad. Pulp & Pap.—Coch. deaer.

Anheuser-Busch—Crane valves. Coch. hp. softener. Ruths accum.

Atmospheric Nit. Corp.—S. K. Valves Perm. zeol. soft.

Bogalusa Paper Co.—Ruths accum.

Champion Fibre Co.—Exhaust to proc. Bled steam to l.p. plant. Coch. st. pur. Perm. zeol. soft.

Chesapeake Corp.—Coch. hp. soft. G. R. evaps. Hagan comb. cont.

De Laval St. Turb. Co.—S. K. valves. Coch. soft.

Dow Chem. Co.—Lime-soda and sulph. acid treat. Cont. blow. S. K. valves.

Du Pont Rayon Co. (Richmond, Va.)—Perm. zeol. soft. & cont. blow.

Fort Howard Paper Co.—Crane valves. P. P. S. C. hp. soft.

Gulf States Steel Co.—Coch. deaer. West. turbs., 15,000 to 4,000 kw.

Lazote, Inc.—S. K. valves.

Mathieson Alkali Co.—Coch. h.p. soft.

Miami Copper Co.—Crane valves. Coch. st. pur.

Minn. & Ontario Paper Co.—Perm. zeol. soft. Coch. deaer.

National Elec. Prod. Co.—Hagan st. pur.

New Haven Pulp & Board Co.—S. K. valves. Foster evap. Coch. deaer.

Ninety-Six Cotton Mills—Perm. zeol. soft.

Penn. Salt Co.—Hagan st. pur.

Pittsburgh Steel Co.—Crane valves. Hagan comb. cont.

Alex. Smith & Sons—Perm. zeol. soft.

H. Smith Paper Co.—Coch. hp. soft.

Firing	Safety Valves	Blow-off Valves	Feed Pumps	Turbines						Feed Reg.
				No.	Make and Type	Kw.	Throttle P. & T.	Bleed, P.	Exh. P.	
P. C. Erie City	Ast. and Cons.	Yarway	Worthington	No						Copes
St., Riley		Yarway	Cameron and Man.	1	West.	7500	385 lb., 660 t	75	12	
				1	G. E.	2500	385 lb., 660 t		7	
	Cons. and Cros.	Yarway	Dean-Hill	1	G. E.	4000	415 lb., 635 t	150	20	
		Yarway		1	G. E.	4000	415 lb., 635 t	150	20	
St., McClave		Yarway	C. H. Wh'lr, Recip.	1	Engine					
P. C. C. E.	Consolidated		DeLaval, Cent.	1	West.	18000	385 lb., 280 s		29-in. vac.	Copes
P. C. Riley	Asht. and Cros.	Yarway	Cameron, Cent.	2	G. E.	300	275 lb.	5	5	
St., Riley			DeLaval, Cent.	2	Allis-Chal.	2000	325 lb., 704 t	Bleeder	28-in. vac.	Copes
St., West		Yarway	Allis-Chal., Cent.	1	Allis-Chal.	2500	300 lb., 160 s	15	28-in. vac.	
Oil	Ashton	Yarway	Worthington	No						
St., Frederick			Worthington	1	West.	1000	285 lb., 200 s		155	
				1	Allis-Chal.	500	300 lb., 622 t		10 lb.	
St. Illinois	Cons. and Cros.	Yarway	Weir		Terry	1000	300 lb., 580 t	15	28-in. vac.	Copes
	Crosby	Yarway	Cent.	2	Stork Bros.		300 lb., 70 s		80	Copes
	Ashton	Yarway								
P. C., Fuller Bonnot	Consolidated	Yarway	Moore	3	Parsons	6000 hp.	325 lb., 710 t	40	5 gage	Copes
P. C., Riley	Asht. and Cons.	Yarway								Copes
St., P. C., Harrington	Consolidated	Yarway	Worhington, Cent.	3	G. E.	1-10000 2-5000 hp.	275 lb., 600 t	9 abs.		Copes
P. C., Fuller Bonnot	Consolidated	Yarway								Copes
	Asht. and Cons.	Yarway	Worth. and Man.							
P. C., Fuller										
St. Taylor	Consolidated		DeLaval, Cent.	1	Nord. Comp. Eng.	1000	350 lb., 600 t		7	Stets
St. Taylor	Consolidated		DeLaval, Cent.	1	DeLaval Turb.	4000	350 lb., 600 t		26-in. vac.	Stets
P. C. Aero	Consolidated		DeLaval, Cent.	1	A. B. B. Turb.	6250	350 lb., 600 t		150	Stets
P. C. Aero	Consolidated		DeLaval, Cent.	1	A. B. B. Turb.	6250	145 lb., 600 t		8	Stets
P. C. Aero	Consolidated		DeLaval, Cent.	1	A. B. B. Turb.	6250	350 lb., 600 t		150	Stets
P. C. Aero	Consolidated		DeLaval, Cent.							
St. U. F.			Triplex	1	Allis-Chal.	500	285 lb., 617 t		15	Copes
				1	Terry	300	300 lb., 560 t	15	28-in. vac.	
Hand Fired	Asht. and Cons.	Yarway	DeLaval, Cent.							Copes
Oil	Consolidated	Yarway								
St., Riley	Consolidated		Worth., Cent.							Copes
St., Coxe	Asht. and Cros.	Eckenroth	Allis-Chal., Cent.	1	Allis-Chal.	12500	255 lb., 525 t	15 abs.-3.7 ab.	29-in. vac.	
St., West		Yarway	Worthington	No						
Wood	Consolidated	Yarway	DeLaval, Cent.	3	2 G. E., 1 West.	7500	375 lb., 550 t	45–140	140 (West.) 29-in.	Copes
St., C. E.		Yarway		2	G. E.	750	300 lb.	30	28½-in. vac.	Copes
St., Riley	Ashton	Yarway	Allis-Chal., Cent.	1	Allis-Chal.	525	250 lb.	35	28-in. vac.	
P. C., Fuller, Bonnot	Consolidated	Yarway	Cameron, Cent.	2	G. E.	4000	275 lb., 100 s	25	28-in. vac.	Copes
P. C., Riley		Yarway	DeLaval, Cent.	2	West.	3000	300 lb., 570 t	20	Cond.	
P. C., C. E.										
	Crosby	Yarway								Copes
St., B. & W. Ch. gr.		Yarway	Worthington, Dup.	2		500	300 lb., 417 t	10	27.9-in. vac.	
St., Taylor	Consolidated	Yarway								Copes
P. C., Fuller Bonnot	Cros. and Cons.	Yarway								Copes
	Consolidated		DeLaval, Cent.	1	West.	500	350 lb., 600 t		125	Copes
St., Taylor	Consolidated		Manistee, Cent.	17	Sturt.		300 lb., 755 t		60 or 10	Copes
St., Taylor	Consolidated	Yarway	Cameron and Penn.	2	G. E.	6000	300 lb., 567 t	85	2, 14 lb. abs.	Copes
St., West.	Consolidated		Goulds, Triplex	1	G. E.	1250	300 lb., 540 t	5	29.5-in. vac.	
P. C., Raymond	Consolidated	Yarway		3	G. E.	5000 & 3000	365 lb., 639 t	Bleed	28.5-in. vac.	Copes
P. C., Riley				2	West.	1000	340 lb., 175 s		30 lb.	
St., Riley	Ashton	Yarway	Allis-Chal., Cent.	1		2500	350 lb., 636 t	10	27-in. vac.	
St., Taylor				1	Terry	200	315 lb., 200 s	5	28-in. vac.	
		Yarway								Copes
	Asht. and Cons.		Allis-Chal., Cent.	1	G. E.	750	325 lb., 550 t			Copes
Oil, S. K.			Worth. Plung.	1	G. E.	125	350 lb.		28-in. vac.	
	Consolidated									
Acid Sludge and Oil	Cons. and Cros.	Yarway	Cameron, Cent.	1	West.	4000	360 lb., 695 t		110	
	Consolidated									
P. C. Riley	Ashton	Yarway	DeLaval, Cent.	2	West.	1250	340 lb., 10 s		30	
P. C. Riley										
P. C., C. E.; St.	Cros. and Asht.									Copes
St., Taylor	Consolidated	Eck. and Yar.	Cameron, Cent.	2	G. E.	2500	300 lb., 640 t	125–25	28-in. vac.	Copes
St., Harrington		Yarway		1	De Laval	300	300 lb., 660 t		25 lb.	
				2	Terry	1000	250 lb., 150 s	40	2 lb.	
St., C. E. Chain gr.		Yarway	Worthington	No						
				1	Moore	750	300 lb., 650 t		20	
			Allis-Chal., Cent.	1	Moore	100				
				1	Terry (paper mach.)	200 hp.	325 lb., 150 s		30 lb.	
Gas or St., Coxe			Worthington, Dup.							
	Consolidated		Mantistee, Cent.	4	West.	15000	275 lb., 575 t		29-in. vac.	
P. C. Aero										
	Consolidated									
Hog Fuel	Crosby	Yarway	DeLaval, Cent.	1	West.	7500	300 lb., 580 t	125	28-in. vac.	
P. C., Riley	Consolidated									

Paul Sorg. Paper Co.—Reiselt lime-barium treat. Wheeler evaps. Coch. deaer.
Standard Oil Co. (Whiting, Ind.)—Crane valves. Perm. zeol. soft.
Toledo Furnace Co.—Perm. zeol. soft.
Vacuum Co.—Perm. zeol. soft. & cont. blow.
Viscose Co.—Perm. zeol. soft. & cont. blow.
Wassau Paper Mills Co.—S. K. valves.
West Va. Pulp & Paper Co.—Hagan st. pur. Perm. zeol. soft. & cont. blow.
Youngstown Sheet & Tube Co.—S. K. valves. Perm. zeol. soft. Hagan comb. cont.

Plants 300 to 399 lb.

Aberfoyle Mfg. Co.—Perm. zeol. soft. & cont. blow.
Bergstrom Paper Co.—Coch. hp. soft. Perm. cont. blow. Hagan comb. cont.
Bryant Paper Co.—Crane valves. Perm. zeol. soft.
Canada Sugar Ref. Co.—Ruths accum.
Central Alloy Steel Corp.—Crane valves. Perm. zeol. soft.
Eagle Paper Co.—P. P. S. C. hp. soft.
Fraser Paper Co.—Crane valves. Coch. deaer.
Glenn Alden Coal Co.—Coch. deaer.
Grays Harbor Pulp & Pepper Co.—Crane valves. Perm. zeol. soft. & cont. blow.
J. Heald Paper Co.—Crane valves. Coch. hp. soft. & deaer. Henzey deconc.
Int. Comb. Tar & Chemical Co.—Perm. zeol. soft.
Minn. & Ontario Paper Co.—Coch. deaer.
National Paper Prod. Co.—Crane valves. Perm. zeol. soft. & cont. blow.
Packard Motor Car Co.—Crane valves.
Rariton Copper Works—Perm. zeol. soft. & cont. blow.
Restigouche Paper Co.—Ruths accum.
Riverside Fibre and Paper Co.—P.S.S.C. hp. soft.
Sunbury Convert. Works—Perm. zeol. soft & cont. blow. Elliott deaer.
Tenn. Coal and Iron Co.—Hagan Comb. Cont.
Thunder Bay Paper Co.—Int. Filt. zeol. soft. Elliott deaer.
Titanium Pigment Co.—S. K. valves. Coch. soft. & st. pur.
U. S. Naval Academy—Perm. zeol. soft.
Virginia Ry. Co.—Perm. zeol. soft. S. K. valves.

Recent High-Pressure Central Stations

In the United States

IN THE accompanying table are summarized the steam conditions of American central stations designed or built since 1924 for steam pressures above 400 lb. These stations fall into three general classes as to steam pressure, namely, 400, 600 and 1,200 lb. in round figures, while they all employ about the same initial steam temperature, ranging from 650 to 750 deg. at the superheater outlet. Within this range of initial steam temperatures the 600-lb. and 1,200-lb. stations require reheat to prevent excessive moisture in the low-pressure turbine stages as well as to better their ideal cycles.

It is interesting to note that of the 41 stations listed, in the next six years will change radically from that obtaining in the past six. It will be interesting to see whether the building of high-pressure plants will practically supplant the building of moderate-pressure plants by 1935.

Available metals have for some time limited the maximum steam temperature to 750 deg. There are now indications that this limit may soon be raised to nearer 1,000 deg. In this event reheating would be unnecessary for plants operating at pressures lower than about 1,000 lb. Thus, if both high temperature and high pressure become available, the pressure range between 650 and 1,000 lb., which is now avoided because of the necessity of reheating, would become available for economic operation. While the use of 1,000 deg. steam temperatures at moderate pressures may now appear possible the combination of high temperature and pressure cannot, be predicted. What effect high temperature will have on future average operating pressures provides a fertile field for those who may wish to venture a guess.

CENTRAL STATIONS OPERATING AT PRESSURES OVER 400 LB.

Station	Location	Operating Co.	Size Unit Kw.	Ultimate Plant Capacity Kw.	Initial Operation Year	Boiler Pressure Lb. per Sq.In.	Boiler Temp. Deg.	Throttle Pressure Lb. per Sq.In.	Throttle Temp. Deg. Fahr.
Avon	Avon, Ohio	Cleveland Electric Illuminating Co	35,000	300,000	1926	400	725	375	700
Avon Park	Avon Park, Fla	Florida Public Service Co	15,000	80,000	1928	448	725	400	715
Binghamton	Binghamton, N. Y.	Binghamton Light, Heat & Power Co	30,000*	85,000	1927	450	725	400	700
Buck	Spencer, N. C	Duke Power Co	35,000	70,000	1926	425	700	350	700
Columbia	Columbia Park, Ohio	Columbia Power Co	63,000	480,000	1925	650	725	600	725
Comal	New Braunfels, Tex	Comal Power Company	40,000	120,000	1926	430	700	...	...
Crawford Ave	Chicago, Ill	Commonwealth Edison Co	104,000*	750,000	1925	580	750	550	725
Deepwater	Deepwater, N. J	American Gas & Electric Co	53,000	400,000	1929	1,200	725	...	...
Delray No. 3	Detroit, Mich	The Detroit Edison Co	50,000	300,000	1929	410	706	375	700
East Peoria	East Peoria, Ill	Illinois Elec. Power Co	23,200*	116,000	1925	400	700	...	...
Edgar	Weymouth, Mass	Edison Elec. Illuminating Co. of Boston	35,000*	350,000	1925	1,400	701	1,200	700
Gould St	Baltimore, Md	Consolidated Gas, Elec. Light & Power Co.	36,000	144,000	1927	450	735	400	715
Grand Avenue	Kansas City, Mo	Kansas City Power & Light Co	10,000	.....	1928	650	740	590	740
Grand Tower	Grand Tower, Ill	Central Illinois Public Service Co	25,000	.....	1923	400	750	360	700
Holtwood	Holtwood, Pa	Pennsylvania Water & Power Co	10,000	120,000	1925	425	560	350	...
Holland	Holland, N. J	New Jersey Power & Light Co	55,000	220,000	1929	1,400	750	1,200	...
Hudson Ave	Brooklyn, N. Y	Brooklyn Edison Co	110,000*	750,000	1927	425	725	375	700
Kalamazoo	Kalamazoo, Mich	Consumers Power Co	20,000	.....	1927	400	700	375	692
Lake Pauline	Quanah, Tex	West Texas Utilities Co	15,000	45,000	1928	400	725	375	700
Lakeside	Milwaukee, Wis	Wisconsin Electric Power Co	37,000*	.....	1927	1,300	...	1,200	730
Lauderdale	Ft. Lauderdale, Fla	Florida Power & Light Co	50,000	150,000	1926	450	750	375	700
Long Beach	Long Beach, Calif	So. California Edison Co	90,000*	750,000	1928	460	750	400	725
Mad River	Springfield, Ohio	Ohio Edison Co	20,000	60,000	1927	450	705	375	692
Morrell St	Detroit, Mich	Detroit Public Lighting Commission	20,000	150,000	1927	400	725	...	...
Neches	Beaumont, Tex	Gulf States Utilities Co	21,000 35,000	200,000	1926 1928	400	700	375	700
Northeast	Kansas City	Kansas City Light & Power Co	45,000*	.....	1928	1,400	725	1,200	725
Philo	Philo, Ohio	American Gas & Electric Co	40,000	240,000	1924	600	700	550	685
Powerton	Pekin, Ill	Super Power Co. of Illinois	52,500	.....	1928	650	725	...	...
Reeves	Norfolk, Va	Virginia Electric & Power Co	30,000	.....	1928	450	725	400	700
Richmond	Philadelphia, Pa	Philadelphia Electric Co	60,000	720,000	1925	420	705	375	675
Riverside	Davenport, Iowa	Riverside Power Manufacture	20,000	150,000	1925	425	700	400	700
South Amboy	South Amboy, N. J	Jersey Central Power & Light Co	25,000	270,000	1929	1,400	750	1,250	750
Stanton	Pittston, Pa	American Gas & Electric Co	50,000	400,000	1927	625	750	600	725
State Line	State Line, Ind	State Line Generating Co	208,000	1,000,000	1929	800	...	600	730
Station A	San Francisco, Calif	Pacific Gas & Electric Co	50,000	250,000	1930	1,400	750	1,200	750
Station C	Oakland, Calif	Pacific Gas & Electric Co	37,500	175,000	1928	450	730	415	700
Toronto	Toronto, Ohio	Pennsylvania Ohio Power & Light Co	35,000	280,000	1925	400	725	350	700
Trenton Channel	Trenton, Mich	The Detroit Edison Co	50,000	300,000	1924	410	706	375	700
Trinidad	Trinidad, Tex	Texas Public Utilities Co	40,000	160,000	1926	450	750	375	700
Twin Branch	Mishawaka, Ind	American Gas & Electric Co	40,000	240,000	1925	650	725	600	725
Waukegan	Waukegan, Ill	Public Service Company of Northern Illinois	50,000*	500,000	1927	635	...	600	725

* Station contains units of a different size or for different steam conditions.

25 are in the 400-lb. class, nine in the 600-lb. class and seven in the 1,200-lb. class. Evidently, the 1,200 lb. advocates are backing their beliefs almost as strongly as those urging 600 lb. as the best all around pressure, while the 400-lb. plants constitute a preponderant majority of the central stations built since 1924. Undoubtedly, the ratio of high-pressure to moderate-pressure plants built

DESIGN OF HOLLAND STATION

Reveals New Solutions Of High-Pressure Problems

HIGH-PRESSURE plants, to generate cheap and dependable power, cannot be designed simply as replicas of low-pressure plants, with boilers, turbines and auxiliaries modified for safe and efficient operation under new and radically changed conditions. While the experience gained with low-pressure equipment should, of course, be utilized as far as possible at every step, the design must be worked out anew from the ground up. Only in this way can full advantage be taken of the fuel-saving possibilities of higher pressures without sacrificing reliability or incurring undue fixed charges.

This point of view is exemplified in the design of the 1,400-lb. station now being built at Holland, New Jersey, on the Delaware River, for the Pennsylvania - New Jersey Power System of the General Gas & Electric Corporation.

The Holland Station, designed and constructed by W. S. Barstow & Company, Inc., will have an ultimate capacity of 220,000 kw. The initial installation, which should go into operation early in 1930, consists of one 55,000-kw. cross-compound turbine-generator unit with an 11,800-kw. high-pressure element and a 43,200-kw. low-pressure element. Steam for this initial installation will be supplied by two horizontal water - tube boilers equipped with superheaters, reheaters, economizers and air heaters.

The principal features of this station have already been described, but the present article will shed additional light on the influence of high pressures in molding the designs, not only of boiler units, feed pumps and turbines, but of such equipment as condensers, fuel burners and soot blowers, which at first sight appear totally unrelated to high-pressure plant design.

Successful solution of any new engineering problem requires an understanding of the specific obstacles to be overcome and the limits that must be adhered to. The best approach to an appreciation of the engineering reasoning back of the design

The designers of the new 1,400-lb. pressure Holland station did not start out with high pressures in mind. Their job, as they saw it, was not to play with fascinating new ideas, but to build a station that would generate reliable power at the lowest possible over-all cost. But, strange to say, the results of extensive and unprejudiced studies forced them to the conclusion that they could not afford, as engineers using business judgment, to build a low-pressure station. If their reasoning is sound, we have entered a new era of high-pressure development.

is to list first the special problems that go with high-pressure operation and then show how they have been solved in the Holland design. The principal special objectives in the design and operation of high-pressure plants may be tabulated as follows:

1. Low first cost—
   (*a*) Low cost of boiler units.
   (*b*) Low cost of heaters and condensing equipment.
   (*c*) Low cost of reheaters and piping.
   (*d*) Low cost of reserve capacity.
2. High efficiency over expected load range—
   (*a*) Flat performance curve for turbine elements.
   (*b*) Proper selection of reheating pressure.
   (*c*) Maintenance of maximum safe reheat temperature at all loads.
   (*d*) Ample use of regenerative heating.
3. Low auxiliary power consumption—
   (*a*) Low power consumption of feed pumps.
   (*b*) Low power consumption of other pumps.
4. Reliability—
   (*a*) Boiler feed water of maximum purity.
   (*b*) Avoidance of tube and water-wall trouble.
   (*c*) Ability to operate with high-pressure turbine shut down.
   (*d*) Avoidance of excess reheat temperature.
   (*e*) Ability to start plant without steam for condenser air ejectors.
   (*f*) Protection of equipment (but not of load) in case of accidental shutdown of station.
5. Intelligent operation without excessive labor cost—
   (*a*) Maximum use of automatic control.
   (*b*) Centralization of control.
   (*c*) Elimination of dust (not specifically a high-pressure problem).

Solution of High-Pressure Problems

Each of these objectives has been carefully considered in the design of Holland Station. The resulting design elements and the engineering reasoning behind them may be outlined briefly as follows:

That the problem of cost had most careful consideration is evident from the fact that the estimated cost of the station is not more than $6 per kilowatt above that of a low-pressure plant designed for maximum commercial economy under identical conditions.

It is the belief of the designers that "base-load" plants are largely a fiction. If history repeats itself, the base-load plants of today will not be so favored a few years hence.

It was therefore decided that the Holland Station must justify itself as a plant to operate at the estimated long-term system capacity factor of 50 per cent, with coal at $5 per ton.

A mark to surpass was set by the low-cost and highly efficient Middletown station near the western end of the system. Middletown, constructed at a total cost of $94 per kilowatt (including all overhead and interest during construction), burns $4.50 coal and operates at 350 lb. pressure.

The designers did not start out with any desire to experiment with high pressures, but were forced into it by extensive studies. These studies showed conclusively that a 1,400-lb. station would give a lower over-all cost per kilowatt-hour than the Middletown plant in spite of the 50-cent difference in coal cost, and would give them a kilowatt-hour cost substantially lower than that obtainable under the conditions to be met at Holland by any design for lower pressure.

The following combinations were analyzed before deciding on the 1,400-lb. pressure (1,200 lb. at turbine throttle):

400-lb. pressure, 2-stage extraction, no reheat.
400-lb. pressure, 3-stage extraction, no reheat.
550-lb. pressure, 4-stage extraction, reheat.
1,200-lb. pressure, 4-stage extraction, reheat.

The fourth extraction stage on the 1,200-lb. pressure was not justified by the small additional reduction in coal consumption, but for a reason to be explained later.

The low cost finally reached has been obtained by careful economic balance at every point. For example, all preconceived notions were set aside in the matter of correct balance in heating surface between boiler proper, economizer and air heaters.

The high cost of high-pressure heating surface made it desirable to keep that portion of the surface at a minimum and work it at a high rate. With this in mind, and also the matter of circulation, it was decided to use the arrangement shown in Fig. 1 with surface apportioned as follows:

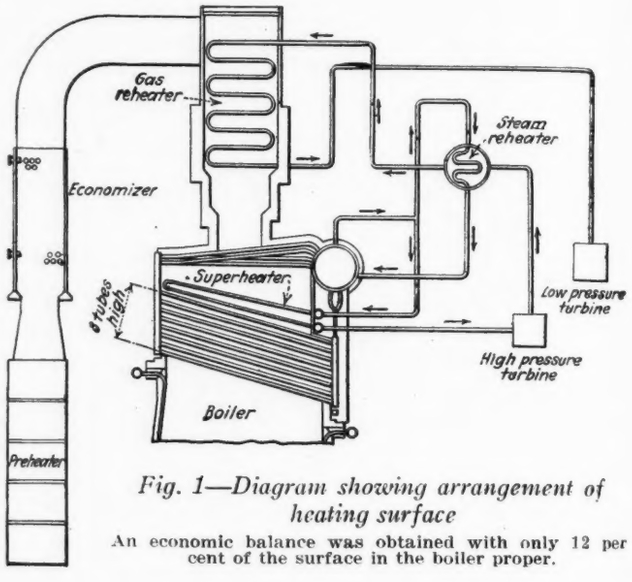


Fig. 1—Diagram showing arrangement of heating surface

An economic balance was obtained with only 12 per cent of the surface in the boiler proper.

Boilers, 12 per cent (7,980 sq.ft.).
Superheater, 6.9 per cent.
Gas reheater, 17.2 per cent.
Economizer, 17.6 per cent.
Air preheater, 46.3 per cent.

It is interesting to note that the boiler surface per kilowatt capacity is only 0.34 sq.ft. as compared with 1.37 sq.ft. per kilowatt with a 400-lb. plant. This explains in part why low station cost is obtainable in spite of the high unit cost of high-pressure boiler heating surface.

Another balancing factor was encountered in connection with heaters and condensers. Increasing extraction stages increased fuel economy and decreased condensing equipment costs slightly, each added heater giving less gain. Balancing of costs on this basis justified three-stage extraction, but not four. A fourth heater was, however, justified as an emergency desuperheater in case of failure of the high-pressure turbine. This will be explained under the head of "reliability."

When isolated central stations are considered, it is of course necessary to install spare prime movers and boiler units, but when many plants are connected by transmission lines the efficient stations are operated and the

less efficient held for reserve capacity in case of outage or emergency in the most efficient stations. Therefore, in building the new plant at Holland, the reserve equipment is considered to be in one of the older stations.

It should be noted, however, that failure of the high-pressure turbine will not put the station out of service, as the low-pressure turbine can be operated alone as a unit and generate 75 per cent of the normal station capacity.

High Efficiency Over Expected Load Range

The next main item is high efficiency. It was found that the available high-pressure turbines did not maintain their efficiency over the wide load range required of a plant designed to share system load fluctuations. To insure a flat performance curve the station designers suggested that the high-pressure turbine element be provided with multi-valves. This suggestion was carried out and resulted in a turbine that maintained its performance over a wide range of loads.

The reheat pressure is allowed to fluctuate from 400 lb. at full load to 83 lb. at quarter load. This permits maximum addition of reheat energy with corresponding improvement in plant economy.

With gas reheat alone, it is necessary either to use special dampers to maintain constant reheat with changing loads or to allow the reheat to fluctuate below the maximum. The latter means considerable loss of plant efficiency. To avoid this, steam reheating is used ahead of gas reheating, with the reheating-steam supply thermostatically controlled to maintain constant temperature in the steam leaving the gas reheater.

Low Auxiliary Power Consumption

Auxiliary drive has a direct effect on over-all station economy. In the Holland design the auxiliaries are completely electrified (with the exception of the emergency boiler-feed pump). Additional savings have been made by grouping the various pumping services (Fig. 2) to avoid waste of developed head.

For example, all equipment served by the general service pumps require approximately the same pressure except the water screens, which use a small amount of water at higher pressure. A small booster pump for this service permits delivery of the main supply at the lower pressure required by the coolers, with a substantial power saving.

Due to cavitation, hotwell pumps ordinarily operate at low efficiency. To avoid imposing this low efficiency on the entire pumping load through No. 2 heater to the primary boiler-feed pump, the hotwell pumps handle the water only as far as the surge tank. Pumping from this point to the intake of the primary boiler feed pump is taken care of by efficient booster pumps.

To avoid the necessity of constructing the third and fourth heaters for boiler-feed pressure, they have been inserted between the primary and secondary feed pumps. The resulting increased temperature of the water handled by the secondary pump substantially increases the load on this pump. The development of high-pressure heaters may perhaps warrant placing two stages of heating beyond the feed pump in future designs, with a reduction in pump load more than enough to justify the increased cost of the heaters.

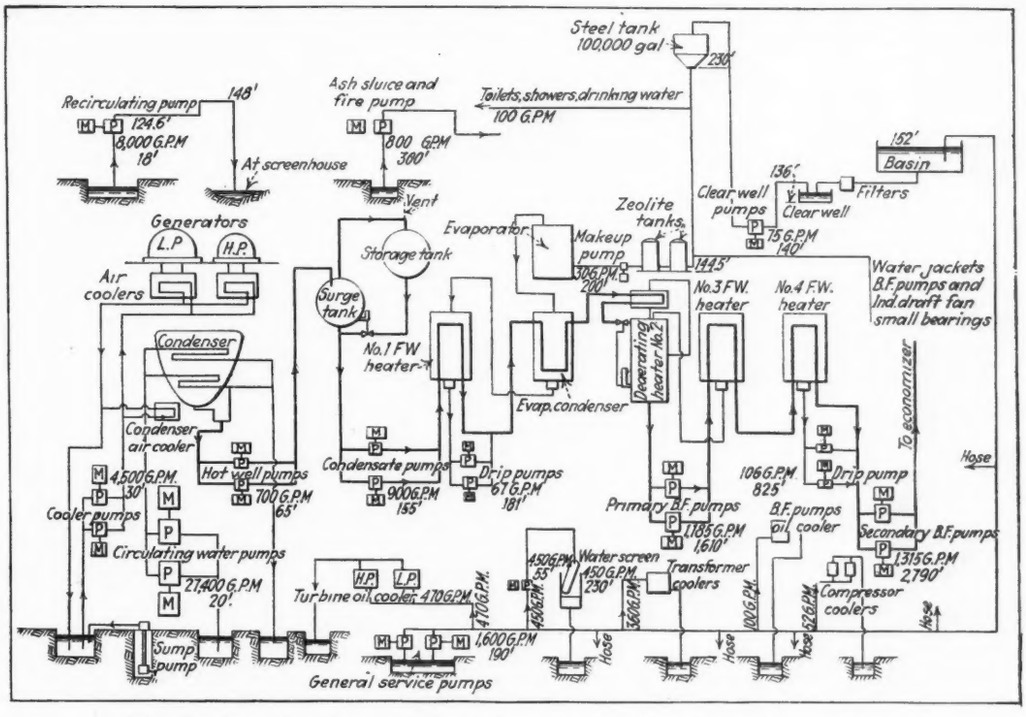


Fig. 2—Pumping power is kept down by intelligent grouping of pumps for various services

How Reliability Is Insured

The problem of reliable operation has been kept constantly in mind. Pure feed water was regarded as a matter of prime importance. It will be insured, in spite of serious contamination of the raw Delaware River water, first, by reducing the makeup water to a minimum and, second, by distilling all makeup from water previously treated by sedimentation, filters and zeolite softeners.

An important reduction in makeup percentages is

secured by the use of compressed air for soot blowing. An ingenious system of gland sealing practically eliminates loss of water from the feed pump or contamination of the feed water.

Again, condenser leakage has been almost entirely eliminated by rolling the tubes in the tube sheets. The water boxes are supported on solid foundations and connected by rubber joints to the shell, which is hung from the turbine and is free to move in relation to the tubes and water boxes.

Some of the trouble experienced with high-pressure boiler units has been due to flame impingement on water-heating surface. To avoid this the Holland boilers are fired with burners opposed in pairs.

Since about three-fourths of the total power is generated in the low-pressure turbine, failure of the high-pressure turbine need not cause a serious curtailment of power production, provided the high-pressure boilers can deliver steam at suitable conditions to the low-pressure turbine. Provision must be made to lower the pressure

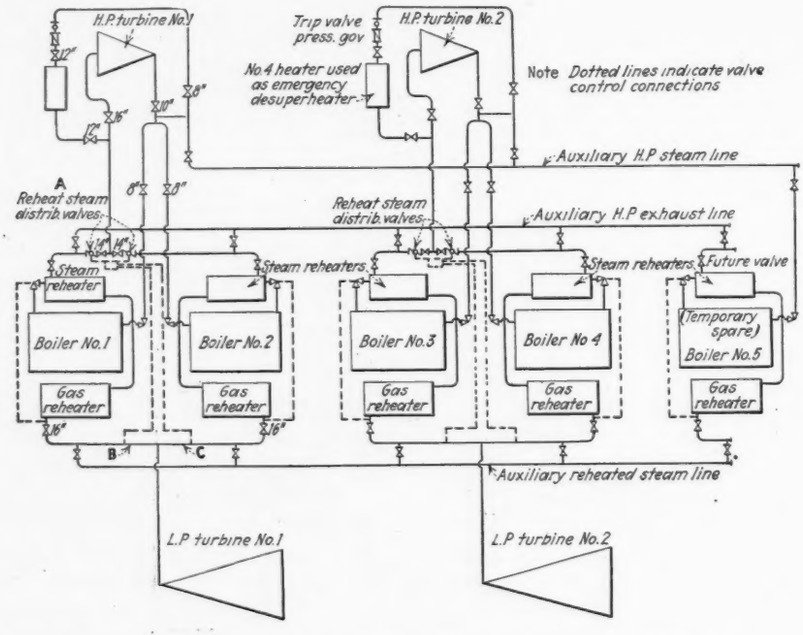


Fig. 3—Diagram of steam distribution between boilers and turbines, showing reheat distribution valves A thermostatically controlled from B and C

of the steam delivered without raising its total temperature. In the Holland Station the shifting over of a few valves will substitute the fourth-stage heater and a reducing valve for the high-pressure turbine element. This heater will then operate automatically as a desuperheater so that the entire boiler system, including steam and gas reheaters, will continue operation without damage or disturbance. This breakdown function of the fourth-stage heater, rather than its slight effect on station economy was the justification for its installation.

The reheat control system is designed to maintain a constant and equal reheat temperature in the steam from both boilers. The method of regulation is indicated diagrammatically in Fig. 3. A special twin valve *A*, controlled by the *difference* between the temperature at *B* and *C*, distributes high-pressure exhaust to the two reheater systems in such a way as to maintain equal temperatures in the steam leaving the two gas reheaters. This steam-distributing device consists of two special balanced valves, motor operated through connected bevel gears in such a way that the opening of one valve by the differential thermostat closes the other an equal amount. In the neutral position both valves are well off their seats. Neither is ever completely closed.

In addition, an individual thermostat on each boiler admits enough live steam to its primary reheater to maintain a constant temperature in the steam leaving the gas reheater.

An unusual feature of this plant, is the combined use of steam-jet air ejectors in conjunction with reciprocating vacuum pumps. These motor-driven pumps are designed to handle a vacuum of 27 in. when no low-pressure steam is available. In normal operation a primary steam-jet air ejector serves to boost the pressure from the 29-in. vacuum in the condenser to the 27-in. at the intake of the reciprocating pump.

Returning to the consideration of normal high-pressure operation, the same reheat regulating devices which serve station economy by keeping the reheat temperature at a fixed maximum under all conditions serve also to prevent damage to equipment by excess temperature.

The final requirement of low-cost reliable power generation with a high-pressure station is a layout that will permit intelligent control without excessive labor cost. This means a maximum use of automatic control devices and a centralization of control of both boiler and turbine room under a single alert and intelligent operator. No wall separates the boiler room and turbine room in the Holland Station. Control is centralized at a point where boilers and turbines are in plain view.

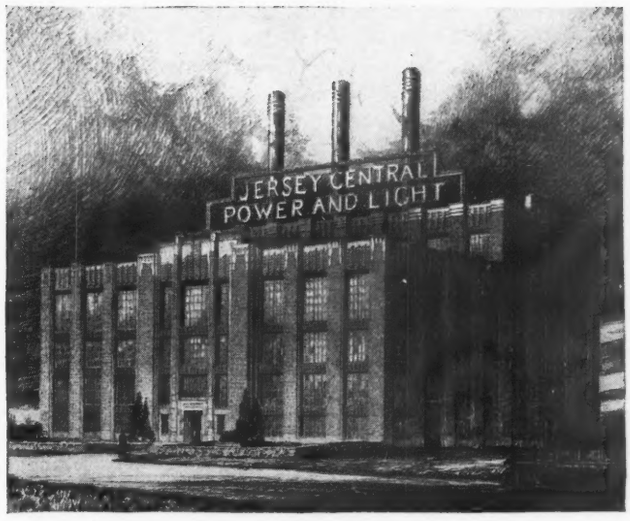
Cleanliness and good operation are closely related. Coal and ash dust in and around Holland Station will be practically eliminated by the burning of coal from the pulverizer cyclone vents in the boiler furnace, and the hydraulic sluicing of ashes.

To sum up, the Holland Station design, by careful consideration of all the factors that influence first cost, operating efficiency and reliability will, it is believed, generate dependable power at a substantially lower total cost than could be obtained with any conceivable design of a low-pressure station burning five-dollar coal operating at the system capacity factor of 50 per cent, and under other local conditions at Holland.

At the present stage of construction there is little chance of any major variation from the estimated station cost and efficiency. If the operating reliability meets expectations, the Holland station may well mark the arrival of a new era in power generation wherein new low-pressure central stations may be hard to justify as a sound investment except where the cost of fuel or the capacity factor is low. If, with the development of high-pressure equipment on a production basis, the slight margin of excess first cost can be erased, even low capacity factor and low fuel cost will cease to be valid arguments against high pressures.

This article was prepared with the co-operation of J. A. Powell, mechanical engineer, and D. C. Hormell, sponsor engineer, W. S. Barstow & Company, Inc.

JERSEY CENTRAL POWER & LIGHT *Builds*



1400-lb. Plant at South Amboy

ANOTHER high-pressure steam-electric station is under construction. Foundation piles have been driven on the Raritan River at South Amboy, N. J., for the 1,400-lb. station of the Jersey Central Power & Light Company.

Each new high-pressure station that goes up seems to incorporate new design elements, a natural result of the application of original thinking to novel conditions. The South Amboy Station is no exception and will embody such innovations as "steeple-compound" turbines, muliple-unit pulverizers with zone combustion control, bowed-tube condensers with the tubes rolled into the sheets, five-stage feed heating, high-pressure desuperheaters and (possibly) heaters of the direct-contact type.

The initial installation will consist of two 25,000-kw. compound turbine-generators, and three (one a spare) boilers with provision for an ultimate capacity of 270,000 kilowatts.

The designers chose 1,400-lb. boiler pressure (1,250-lb. at the turbine) for strictly commercial reasons after exhaustive studies had shown lower-pressure plants to be inferior investments under the existing conditions. The following proposals were analyzed:

(A) 400 to 450 lb. pressure;
two single-cylinder 25,000-kw. turbines;
three 16,360-sq.ft., 3-pass boilers, with superheaters and large air heaters; no reheat;
three-point bleeder heating.

(B) 700 to 770 lb. pressure;
two tandem compound 25,000-kw. turbines;
three 7,250-sq.ft. boilers with superheaters, gas reheaters, economizers and air heaters;
four-point bleeder heating.

(C) 1,200 to 1,400 lb. pressure;
two cross-compound 25,000-kw. turbines;
three 7,250-sq.ft. boilers with superheaters, steam and gas reheaters, economizers and air heaters;
four- and five-point bleeder heating.

Similar studies were made for a 40,000-kw. installation consisting of two 20,000-kw. units, but the cost per kilowatt ran so much higher than that for two 25,000-kw. units that the 40,000-kw. installation could not be seriously considered.

The estimated economy with the various turbines is shown in Fig. 1, 1,200 lb. being further analyzed for the effect of constant superheat, constant reheat and a multiple-valve high-pressure turbine with flat efficiency curves.

In each case the assumed throttle temperature was 750 deg. at full load, which was likewise the reheat temperature in plants *B* and *C*. Studies were made for the highest pressure to show the effect of allowing the superheat and reheat to vary naturally with the load and the improvement when these temperatures, one or both, were kept constant.

Estimated Station Cost Increases Slightly with Pressure

Taking the 400- to 450-lb. plant as a basis, the additional cost of the 700- to 770-lb. installation was estimated at $6.75 per kilowatt and of the 1,200- to 1,400-lb. installation at $10.75. These figures differ from those obtained by other designers of high-pressure stations in that the 1,200- to 1,400-lb. station costs substantially more than the 700- to 770-lb. station. Others have estimated the cost of a 1,200-lb. station as equal to or slightly lower than that of a 700-lb. plant. It should be recognized in this connection that the relative costs vary according to the layout assumed for each pressure. A substantial part of the difference between the cost of 700- to 770-lb. and 1,200- to 1,400-lb. plants in this case is due to the fact that the boilers of the former use riveted drums, while the drums of the 1,200- to 1,400-lb. boilers are seamless forgings as in other American plants of this pressure.

Balancing cost against fuel saving, the 700-lb. to

770-lb. station was shown to be definitely superior to the 400-lb. to 450-lb. station, while the 1,200-lb. to 1,400-lb. station, in turn, showed a definite commercial advantage over the 700-lb. to 770-lb. plant.

This left no choice other than 1,200- to 1,400-lb. if assurance could be had that a plant built for this pressure would be thoroughly dependable and easy to

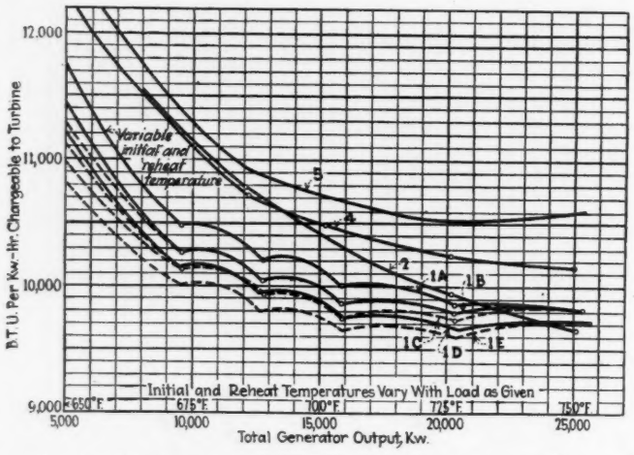


Fig. 1—Heat rate comparisons for various conditions with 25,000-kw. units

1—1,200-lb. pressure. Cross-compound turbine with 5 control valves.
A—Variable initial and reheat temperature.
B—Variable initial temperature. Reheat temperature constant at 750 deg.
C—Initial and reheat temperature constant at 750 deg.
D—5 bleeds. Variable initial temperature. Reheat temperature constant at 750 deg.
E—5 bleeds. Initial and reheat temperature constant at 750 deg.
2—Same as 1, except one control valve.
4—700-lb. initial pressure. Tandem compound turb., multivalve.
5—700-lb. initial pressure. Variable initial temperature. Steam reheat to 500 deg. (constant). Max. reheat pressure 20 lb.

operate. A careful study of the operating experience of existing high-pressure plants led the designers to the conclusion that high-pressure installations had passed the experimental stage and that dependable operation could be counted on. It was found that at least twelve boilers would be in operation at these pressures by next summer and that one plant had operated for four years at 1,200-lb. pressure. A study of troubles experienced showed that all were of a minor nature and that practically none was due intrinsically to high-pressure operation.

As to ease of operation, it was concluded that the 1,200-lb. installation would be as easy to control as a 700-lb. plant and nearly equivalent to a 400-lb. plant.

Another factor in the choice of 1,200- to 1,400-lb. was the desire to avoid the early obsolescence that threatens low-pressure stations erected at the present time. In short, it was felt that a high-pressure plant was a distinctly better investment from the point of view of security as well as rate of return.

One incidental result of the erection of high-pressure plants will be to speed the death of "boiler horsepower." This anachronism, which once bore some faint relation to the engine horsepower that could be supplied by a boiler, becomes a truly ridiculous measure in the modern 1,200-lb. plant.

At South Amboy two 721-"horsepower" boilers (7,214 sq.ft. each), a total of 1,442 "horsepower," will supply steam to generate 50,000 kw., or 34 kw. per "horsepower." With 0.29 sq.ft. of heating surface per kilowatt, the boiler seems to be almost a vanishing element. This is largely the secret of the surprisingly low cost of high-pressure plants in the face of the high unit cost of the heating surfacing in forged-drum boilers.

The heating surface in each of the South Amboy boiler units is apportioned as follows:

	Sq.Ft.
Boiler	7,214
Superheater	5,900
Desuperheater	225
Gas reheater	12,900
Steam reheater	2,269
Air heater	32,052
Economizer	4,423
Total	64,983

It will be noticed that only 11 per cent of the heating surface is in the boiler proper.

All these elements are of Babcock & Wilcox manufacture. Their arrangement is shown in the boiler cross-section (Fig. 3). An interesting feature of this arrangement is the placing of the steam reheater directly above the boiler, where the condensate can drain into the boiler drum. A further point of considerable interest is the arrangement for control of steam temperature. The reheat temperature is thermostatically controlled by varying the quantity of steam admitted to the steam reheater, and the initial steam temperature is thermostatically controlled by varying the water level in the desuperheater, which is of tubular type.

Each boiler will be fired through five burners by five unit pulverizers with individual primary air fans. Secondary air for each boiler will be supplied by two fans and subdivided and accurately measured for each burner. The pulverizers are 5,000-lb. per hour ball mills manufactured by Kennedy Van Sain Mfg. & Eng. Corp., with Fuller Lehigh burners and furnaces.

With this system of firing will go an entirely new type of combustion control now being perfected by the station designers in co-operation with the Bailey Meter Company and the Toledo Scale Company. Details are not at present available, but it is announced that this system will proportion the air and coal individually to each burner to maintain the proper fuel-air ratio in all parts of the furnace. A very high efficiency is expected as a result of this arrangement.

Other details of the design are still incomplete, but sufficiently advanced to indicate developments of great interest. For example, in the five-step feed heating sys-

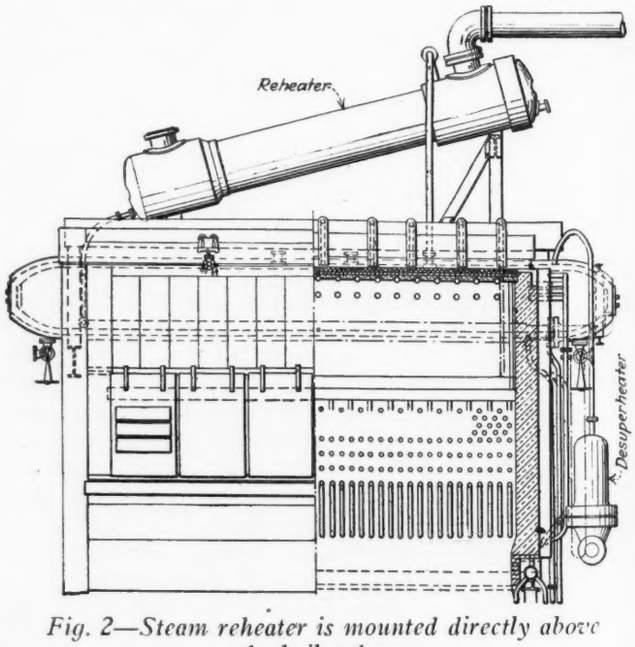


Fig. 2—Steam reheater is mounted directly above the boiler drum

tem adopted, serious consideration is being given to the elimination of tubular heaters. It is proposed to replace these by direct contact or "open" heaters, thereby doing away with the temperature drop in tubular heaters and all the complication and loss of cascading and drip pumping.

A novel feature of the South Amboy plant will be the "steeple" arrangement of the cross-compound General Electric turbines. A substantial saving in floor space and piping is accomplished by the bold step of mounting the high-pressure turbine and generator directly on top of the low-pressure generator, as shown in Fig. 4.

The capacity of the complete cross-compound unit is 25,000 kw., 5,700 kw. in the high-pressure cylinder and 19,300 kw. in the low-pressure cylinder. The relative size of the high-pressure unit is even less than might be expected from this capacity relation, since it operates at 3,600 r.p.m., as against 1,800 r.p.m. for the low-pressure unit.

Current is generated at 13,200 volts, three-phase, 60-cycle. A direct-connected exciter is placed in the low-pressure generator, with one motor-generator set per unit as a stand-by.

Guarantees received on the 1,250-lb. turbine indicate a station efficiency somewhat higher than the estimate used in comparing the 400-, 700- and 1,200-lb. installations, and computing the curves shown in Fig. 1.

As in the Holland Station, makeup is kept at a minimum by rolling in the condenser tubes and by using compressed air instead of steam for soot blowing. Unlike Holland, the condenser shell is a unit, expansion of the tubes being provided for by initial bowing. This is a 20,000-sq.ft. Foster-Wheeler single-pass condenser with the tubes arranged radially.

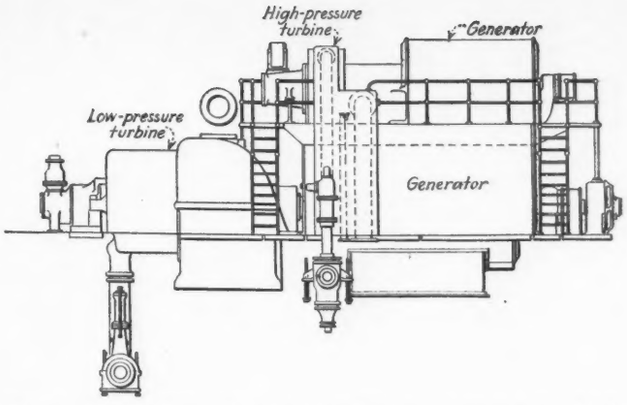


Fig. 4—In the "steeple-compound" arrangement the high-pressure unit is mounted on the low-pressure generator

While, at the present writing the details of condensate pumps and feed pumps have not yet been settled, quintuple reciprocating feed pumps have been considered. These show a substantial saving in power because of their much higher efficiency, but the centrifugal pumps have been tentatively selected because of their much lower cost and space requirements.

They will be special pumps designed for variable-speed operation from the condensate to the pressure of the fifth bleed point and constant-speed operation thence to the boiler-feed point, synchronized by common driving mechanism and common control mechanism to fit properly the conditions they must work against. This proposed arrangement increases the efficiency of the boiler feed and condensate pumping, decreases the number of pumps required and works in properly with the proposed heating system.

One 7,050-lb.-per-hour evaporator is provided for each unit. This takes steam from the No. 3 bleed point.

All auxiliaries in the South Amboy Station will be motor driven, with a spare condensate pump, circulating pump and boiler feed pump for each unit.

It is expected that the plant will be ready to operate in the early summer of 1930.

This preliminary article has been prepared with the co-operation of R. C. Roe, engineer in charge of design and erection of the South Amboy plant for the Electric Management & Engineering Corporation, New York City, the designers and constructors. Shreve & Lamb are consulting architects. Sargent & Lundy of Chicago, Illinois, are advisory engineers.

Credit is also due to the various equipment manufacturers for splendid co-operation and engineering help on this project.

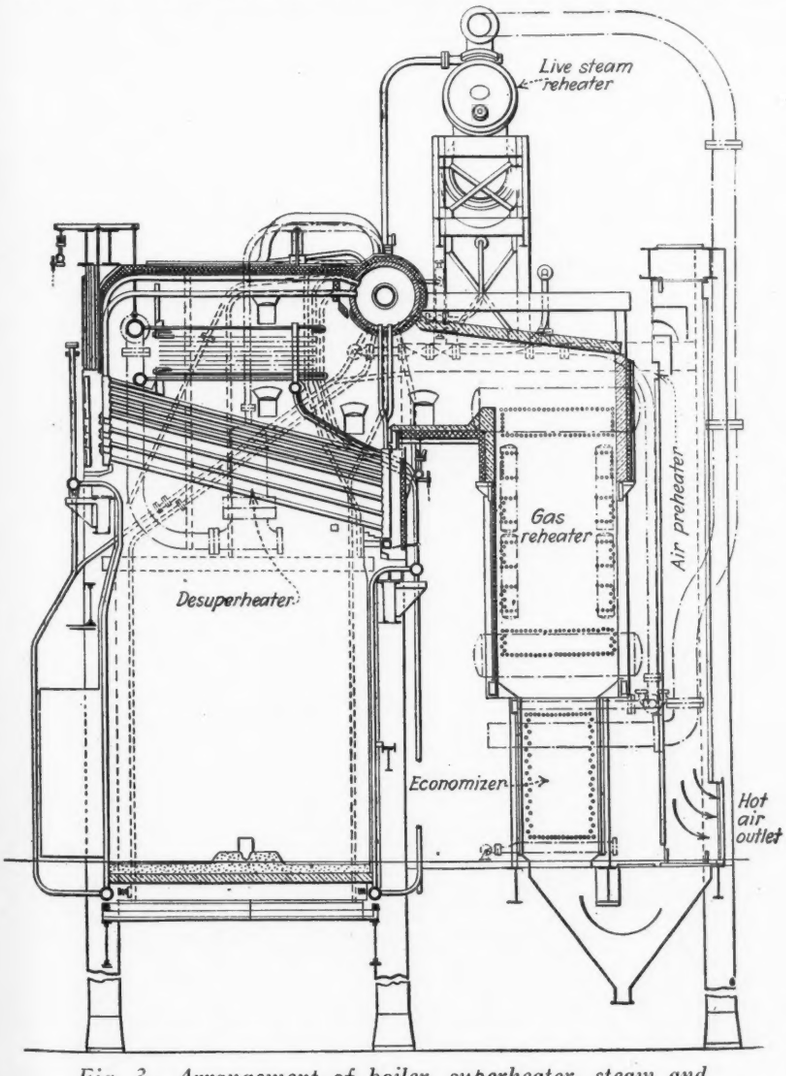


Fig. 3—Arrangement of boiler, superheater, steam and gas reheaters, economizer and air preheater

HIGH PRESSURES
Their Effect on
BOILER DESIGN

Forged drums, different apportioning of heating surfaces and greater attention to circulation are some requirements of pressures over 1,000 pounds

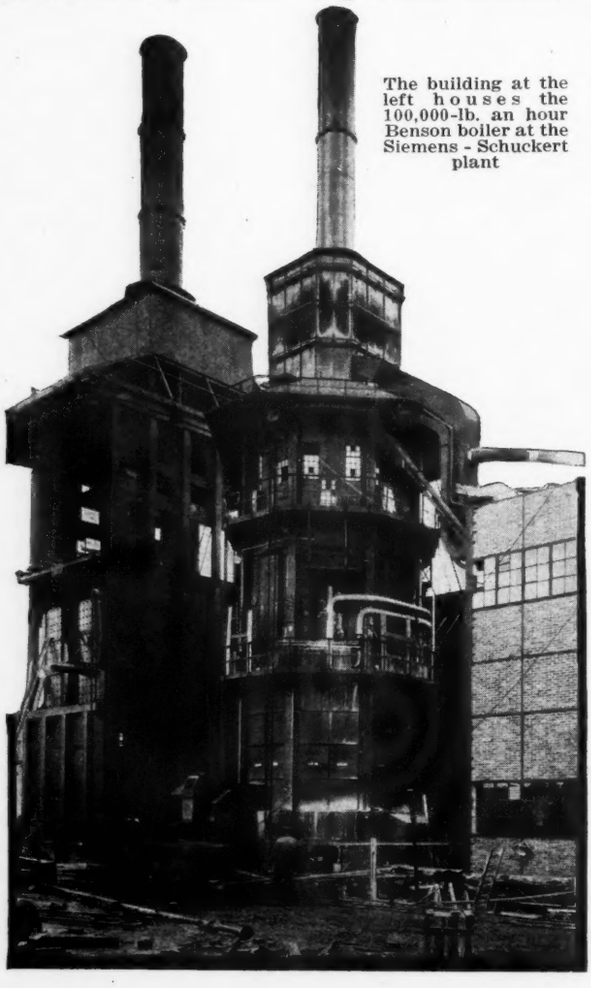
By J. B. Crane
High-Pressure Specialist, Combustion Engineering Corporation

TODAY there are in operation in the United States one plant at 800 lb. pressure, five plants at 1,400 lb. pressure, three plants building for 1,400 lb. pressure and one plant building for 1,800 lb. pressure. Of the five plants operating at 1,400 lb., three have ordered additional equipment.

Their operating record has been such as to indicate that troubles have been less than expected and that materials are obtainable and engineering talent is available to cope successfully with such problems as are presented.

In Europe the problem of generating at high pressures has been attacked a little differently. Materials there are about 60 per cent of the cost here, and consequently it has been possible to build small units for test purposes, while here such units were so expensive that it seemed better to construct units large enough to become integral parts of large generating stations. Consequently the designs have rather closely followed those that have been found to work well at lower pressure, and the increase in pressure has been gradual. European engineers have followed the same general designs as here for large units, but some have tried to omit drums and use tubes, thus cutting down first cost. The three main attempts along this line are the Atmos, Benson and Loeffler boilers.

The Atmos boiler was developed in Sweden and consists of rotating tubes which secure a high heat transfer per square foot. The original boiler contained 13 sq.ft. of heating surface and consisted of four tubes 12 in. outside diameter with 0.670-in. walls and 11-ft. effective length in the furnace. This boiler generated, it is claimed, 10,000 lb. of steam per hour at 1,400 lb. pressure and 840 deg. steam temperature. A French company took over the license for this boiler and has erected two units for experimental work, one of which is similar to the original boiler, but the second has a squirrel-cage rotor. With this new arrangement the ordinary rotor is surrounded by an outer ring of tubes connected at the front and rear to the main rotor by tubes. The speed of this rotor is 20 r.p.m. as against 330 r.p.m. on the old rotors. The power is reduced, and it is claimed that a single rotor output of 12,000 lb. of steam can be obtained. The stuffing boxes are water-cooled and it is said the sealing is perfect. It is reported that they are now building a 25,000-lb. unit for 1,400 lb. pressure for installation in an industrial plant. As we have not considered any unit of this pressure for less than 100,000 lb. of steam per hour, and as our engineers are likely to look with disfavor on rotating tubes for this pressure, it is unlikely that we shall use this type of boiler in this country unless it can be built for less money than other forms.



The building at the left houses the 100,000-lb. an hour Benson boiler at the Siemens - Schuckert plant

Benson generates steam at the critical pressure, 3,300 lb., and as there is no boiling, the boiler requires no drums. The water is introduced into one end of a long tube surrounding the furnace and comes out of the other end of the tube as steam. The Siemens-Schuckert plant at

Berlin has had in operation for the last two years a boiler of this type delivering 66,000 lb. of steam per hour at 3,200 lb.

A new unit for 100,000 lb. of steam per hour and 3,300 lb. pressure is being installed and it is expected will be in operation in 1929. It is claimed this cycle is 15 per cent more efficient than the 450 lb. pressure cycle, while the 1,200 lb. pressure cycle is 13 per cent more efficient, but because of the lack of drums, it is less expensive to install than the 1,200 lb. pressure.

Loeffler[1] also places coils about the furnace but passes steam through these coils instead of water. The superheated steam passes through coils in drums containing water. The heat is then transferred to the water and in this manner the drums are outside of the setting and can be made of less thickness for the same pressure, there being no tube ligaments to be considered. A special pump is required to compress the steam from the drums into the superheater. An experimental plant for 1,400 lb. pressure and 4,000 lb. of steam per hour has been in operation for three years, and one for the same pressure and an output of 60,000 lb. of steam per hour has been in operation over a year. Larger plants are projected.

As mentioned before, the high-pressure boiler designers here have followed the designs at lower pressures, and where the drums were too thick for plate and rivet construction, they have been made of forgings. The machinery used during the war for making cannons has been put to good use manufacturing boiler drums, oil stills and high-pressure vessels for other purposes.

Both the vertical and the horizontal types of boilers have been used. In general it may be said that the horizontal type is somewhat cheaper on account of the cost of forged drums, but there may be cases where the space limitations in old buildings are such as to make the vertical or semi-vertical type the best choice. The arrangement of superheater, reheater and type of firing will often affect the type of boiler to be chosen. This same selection of boilers is often used abroad, although the lower cost of forged drums in Europe has caused them to favor the semi-vertical boiler, and the largest installations there are of this construction.

The following matters have to be considered in the design of all boilers, but a more intensive study is required in the case of pressures of 600 lb. and above, because we do not have the background of experience that we have for the lower pressures, and, as mentioned before, the demand is for large units so that we cannot stop to experiment with small ones.

The first thing to determine is the size of furnace required to burn the fuel available. If the fuel is low grade, requiring some form of stoker, the total steam requirements must be divided into enough units to secure the proper grate area. If a higher grade of fuel is available, fewer units can be used. The size of furnace to burn the required amount of fuel without slagging and with a minimum of carbon loss is then determined. The proper amount of convection surface that will evaporate the required amount of steam and give the requisite protection to the superheater is then determined.

The air preheater having the cheapest surface, the largest air preheater, considering the space available, is next chosen, working back from the exit temperature of the gases for the efficiency desired. If there is any difference between the entering temperature to the air heater and the temperature leaving the superheater, this gap is made up by installing additional convection surface in the boiler, if possible, or by the installation of an economizer or by both. In order to make the high-pressure surface as low as possible in first cost, it is usually desired to operate at high rates of evaporation for the different surfaces and to install water walls and all other accessories for high ratings.

By making certain assumptions about boiler circulation it is possible to approximate actual operating conditions, and certainly to anticipate troubles that might otherwise cause failure. These assumptions are:

1. Circulation once established remains the same, even if the heating medium is placed elsewhere. This fact has been determined by test, and, while operating conditions may change this circulation, it will not interfere with correct operation.

2. The resistance of the tubes to flow with rising water and steam must equal the resistance to flow of the downcoming water and steam, if the temperature of gas causes any steam to form. We then pick the tubes nearest the fire as risers, determine the resistance to flow with varying amounts of water and plot a curve. We then take the downcomers and plot a similar curve. Where these curves cross will be the actual flow taking place. This is then divided by the total evaporation of the boiler, which determines the number of times per hour that the water and steam circulates in the tubes. With this information it is possible to calculate the proportion of steam and water in each tube, which can be checked with the B.t.u. absorption of the various surfaces. This information is fairly well known. In this manner a boiler for 250 lb. pressure will show a circulation of fifty times an hour and for 2,000 lb. pressure will show a circulation of only thirteen times per hour. We then study the different parts of the boiler in a similar manner until all parts are carefully checked. Designs so studied have been remarkably free from trouble. It should be remembered that while the circulation is slower and the volume of steam less at high pressures than at low pressures, yet sufficient risers and downcomers must be supplied to take care of the lower pressures existing while starting up and shutting down or trouble will be experienced.

It has been found for high pressures that feed water must be distilled and deaërated, and consequently our problem is not complicated with treated water with its priming and foaming. The difficulty in securing dry steam experienced with low-pressure boilers has not yet confronted us. At higher ratings this point will require serious study.

Because the pressure drops are relatively small at the lower pressure, engineers have often overlooked the necessity for sufficient leeway between the prime mover and the boiler. Take a typical case for a 1,200 lb. pressure turbine.

Pressure at turbine	1,200 lb.
An economical pipe line—10 per cent loss	120
Valves and fittings	20
Superheater	35
Blowdown for safety valves, 4 per cent	40
	1,415

The safety-valve blowdown has been taken as an operating allowance, since the boiler pressure must be lowered by this amount before the valves will close, and unless this allowance is made the pressure at the turbine must drop below the normal operating pressure. In case the safety valves opened when full load was being carried the drop in pressure at the turbine would cause a drop in load that might be embarrassing.

[1] See *Power*, March 9, 1929.

For ordinary rolled and pressed drums the inner radius of the vessel multiplied by the pressure and divided by the allowed stress of the material used determines the thickness of the shell when increased by the seam efficiency or tube ligament efficiency, whichever gives the thicker shell. This applies to shells up to 2 or $2\frac{3}{8}$ in. thick for the usual diameters used for boiler drums. The limit in pressure for this class of work is 600 lb. to 700 lb. pressure.

For thicker drums the problem becomes more complicated. D. K. Clark shows that in a thick cylinder, when P = lb. per sq.in. pressure; S = actual stress in lb. per sq.in.; R = hyperbolic log of the ratio of r' to r; r' = outside radius; r = inside radius; $S = \frac{P}{R}$; t = shell thickness.

Lame for hydraulic work adopts the formula

$$S = P\,\frac{(r'^2 + r^2)}{(r'^2 - r^2)}$$

The A.S.M.E. code has endeavored to take care of thick cylinders by saying that "if the thickness exceeds 10 per cent of the radius, the outside radius must be taken in figuring the thickness" in the following formula:

$$S = \frac{r' \times P}{t}$$

A comparison of these three formulas is shown in the following table calculated for drums of 50 in. outside diameter, pressure as shown and 3, 4, and 5 in. thick.

Pressure	Stress t = 3 In.			Stress t = 4 in.			Stress t = 5 In.		
	Clark	Lame	A.S.M.E.	Clark	Lame	A.S.M.E.	Clark	Lame	A.S.M.E.
800	6,750	6,320	6,666	4,840	4,624	5,000	3,580	3,648	4,000
1,200	10,150	9,780	10,000	7,220	6,936	7,500	5,400	5,472	6,000
1,600	13,500	12,640	13,333	9,650	9,248	10,000	7,150	7,296	8,000
2,000	17,000	15,800	16,666	12,150	11,560	12,500	9,000	9,120	10,000

At the present time this disparity does not make a great deal of difference in the usual run of cases except where a little more thickness requires the next size ingot or throws the ingot into larger furnaces and lathes.

Research work is being conducted by numerous agencies and eventually reliable figures will be available. As soon as we begin using the chrome and nickel alloys this may become more important, but the A.S.M.E. code is amply safe.

When it comes to establishing the safe stress to use, there is a much wider discrepancy. The A.S.M.E. code provides for a factor of safety of five based on the tensile strength. Nothing is said about the temperature at which the tensile strength is to be measured, and at the lower pressures this makes no difference. There seems to be a tendency on the part of other engineers to reduce the factor of safety on the thicker drums, but I believe this is a wrong move until more information is available on creep and other undetermined stresses.

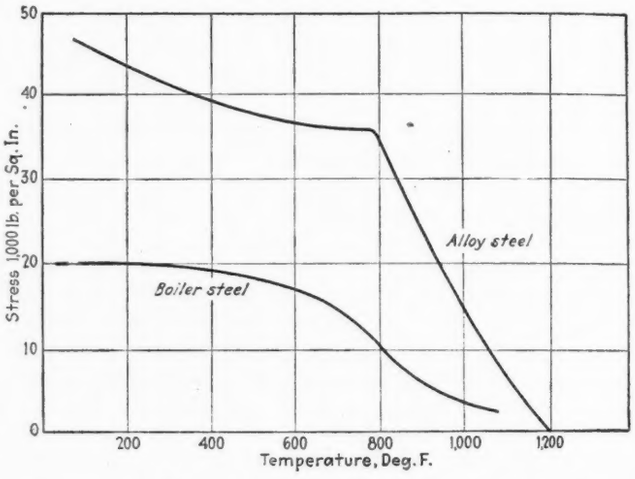
A few tests are available on creep stresses of some steels. These tests number in the hundreds, while they must be in the thousands before definite figures are available. Tests are now made in laboratories on small samples that may represent minimum, average or maximum conditions, and, furthermore, stresses are encountered in actual boiler operation that cannot be duplicated in laboratory work, although these tests give the trend of what may be expected.

In actual practice more or less indeterminate stresses are caused by:

1. Rolling in tubes, since for high pressures considerable force is necessary to make them tight.
2. Beam action of the drums, which can be calculated quite closely, but is often neglected.
3. Distortion, encountered when starting up and shutting down a boiler, and due to unequal temperatures in various sections of the drums. This may, and usually does, cause a hump of two inches in the center of a long drum, which disappears when full pressure and temperature is reached.
4. Unequal expansion due to the gas side of the drum being hotter than the side outside of the boiler setting. This occurs even when the drum is protected from the direct impingement of the hot gases by refractories or insulation, as these become heated through and expansion stresses are set up
5. Other stresses are set up due to surges, secondary combustion or low water and there may be minor or major explosions.

For these reasons it is suggested that a permanent record be kept of annual readings of length, diameter, circumference and sag. This, if plotted, will give a life history of each drum, so that trouble will be discernible and corrected before serious damage occurs.

French, in the Bureau of Standards paper No. 362,



Stress required to produce 0.1 per cent elongation in 1,000 hours

gives the values shown in the illustration for two different metals. These are on the basis of stress producing 0.1 per cent or less elongation, in 1,000 hours. The widely different performance should be noted. On the basis of 0.1 per cent elongation in 1,000 hours we would get 0.6 per cent elongation in a year's operation of 6,000 hours, or 6 per cent in ten years. In other words, if we started with a 50-in. drum, it would be 53 in. in diameter at the end of ten years, the metal would be stretched thinner, the stresses would be increased, and the life of the drum would be about reached.

Experiments are under way to eliminate the drum in boilers operating below the critical pressure. This is complicated by the difficulty of getting separation of the steam and water in tubes or headers. The savings to be made by the use of high-pressure steam are enough to warrant the use of forged drums, even if a suitable drumless boiler cannot be developed. It is safe to say that our manufacturers will be able to reduce the cost of forgings as soon as the business can be put on a production basis. Thus far, each job has been special and wanted in a hurry, which is not conducive to low costs of construction.

Design of Superheaters

For High Pressures and High Temperatures

By R. M. Gates and C. W. Gordon
Vice-President *Mechanical Engineer*
The Superheater Company

IN EUROPE the trend in central-station design is toward moderately high pressures and high initial temperatures. The initial temperature is set at a value sufficiently high to eliminate reheating if possible. If, however, it is necessary to resort to reheating, steam reheaters are favored.

On the Continent final temperatures of 800 deg. F. and above are quite common. The high-pressure, high-temperature section of the Langerbrugge Station, of the Centrales Electriques des Flandres, has been in successful operation since 1925 at a pressure of 800 lb. per sq.in. and a final temperature of 840 deg. F.

In England several stations are in operation or projected, at final steam temperatures ranging from 775 to 840 deg. F. The Battersea Station of the London Power Company will operate at 645 lb. per sq.in. and 875 deg. F. final steam temperature.

In the United States the trend is toward high pressures and moderately high temperatures. Thus, we have several stations in operation at a pressure of 1,400 lb. per sq.in. and final steam temperatures of 700 to 750 deg. F. (750 deg. F. represents the maximum final steam temperature in use at the present time in this country). At least one station however, is projected for a final steam temperature of 775 deg. F.

It is possible to obtain the same thermal efficiency with moderately high pressures and high temperatures, or with high pressures and moderately high temperatures. By judiciously using the materials now available for superheater construction, superheaters may be designed for a final steam temperature of 1,000 deg. F. and a pressure of at least 850 lb. per sq.in. This can be accomplished without the use of special alloy tubing. When special material is used it is possible to design superheaters for a final steam temperature of 1,000 deg. F. and pressures of at least 1,400 lb. per sq.in. Thus, superheater design does not present any obstacles to the employment of high steam temperatures at moderately high and high pressures.

PRINCIPLES of design that permit using the commonly available materials in superheaters for high pressures and moderately high temperatures (1,400 lb. per sq.in. and 800 deg. F. final steam temperature).

* * *

The use of low-carbon steel surface-treated tubing permits the design of superheaters for very high steam temperatures and moderate pressures (900 lb. per sq.in. and 1,000 deg. F. final steam temperature).

* * *

For the combination of very high pressures and very high temperatures special alloy tubing must be employed (1,400 lb. per sq.in. and 1,000 deg. F. final steam temperature).

* * *

For high-pressure and high-temperature conditions every detail of design must be carefully studied to insure low maintenance charges and maximum continuity of service.

* * *

The use of adequate steam velocities and perfect steam distribution materially extends the temperature limits of commonly available materials.

Materials Available

In the design of superheaters the part subjected to the severest service is the tubing. There are available the following steels:

1. Low-carbon, or mild steel, with a carbon content averaging 0.15.
2. Medium-carbon steel, moderately hard, with a carbon content averaging from 0.30 to 0.40.
3. High-carbon, or hard steel, with a carbon content of 0.50 and above.
4. Carbon steel calorized. Any of these four steels may be successfully calorized, although the process is most extensively applied to low-carbon steel.
5. Chrome molybdenum steel.
6. Chrome nickel steel.

The prices of these steels are approximately in the order listed. Low-carbon steel is most extensively used, as it is readily worked and has

a sufficiently good temperature characteristic to satisfy present demands. The curves in Fig. 1 show recommended design stresses expressed in pounds per square inch plotted against metal temperature in degrees Fahrenheit. These figures may be considered somewhat conservative as they are considerably lower than the safe working stresses for a life of 100,000 hours, as determined by reliable investigators. In computing tube wall thicknesses Barlow's formula should be used. Assuming that the tube-wall temperature is not more than 100 deg. F. higher than the contained steam temperature, it is recommended that the following temperature limits be established for the materials indicated by the curves:

1. Low-carbon steel—Steam temperature, 800 deg. F.; wall temperature, 900 deg. F.
2. Medium-carbon steel—Steam temperature, 850 deg. F.; wall temperature, 950 deg. F.
3. Calorized steel—Steam temperature, 1,000 deg. F.; wall temperature, 1,100 deg. F.
4. Chrome nickel steel—Steam temperature, 1,100 deg. F.; wall temperature, 1,200 deg. F.

Superheaters at Langerbrugge

The superheaters of the Langerbrugge Station are constructed of tubes with an outside diameter of 1½ in. The tubing is of medium-carbon steel with a temperature characteristic corresponding to that shown by the curve in Fig. 1. Information as to the gage of tubing used is not available. It is safe to say, however, that the working unit stress is not greater than that given for this curve at the working temperature of the metal.

The Löffler steam generator uses high-carbon, or hard tubing, for a final steam temperature of 930 deg. F. Information as to the exact working unit stresses is not available at this time. It should be remembered in this case that the steam is pumped through the superheater at a very high velocity.

Calorized tubing has been extensively used in oil-still construction, both for tubes containing oil and for

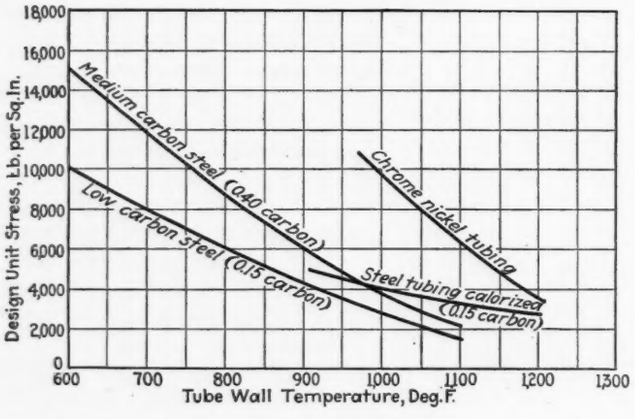


Fig. 1—Unit stresses versus wall temperatures for different materials

superheater tubes. The Calorizing Company reports one successful survey of a superheater operating at 300 lb. per sq.in. pressure and 1,200 deg. F. final steam temperature. The working unit stresses under these conditions were probably 1,800 lb. per sq.in. The same company reports another successful case of calorized tubes used in an oil vapor heater at 125 lb. per sq.in. pressure and 1,100 deg. F. final vapor temperature. A number of oil vapor heaters have been constructed for a working pressure of 50 lb. per sq.in. and a final temperature of 1,150 to 1,200 deg. F.

In setting the temperature limits for a given material, several factors in addition to the limiting creep stress must be taken into consideration:

1. Maximum metal temperature under normal conditions of operation.
2. Maximum metal temperature under starting-up conditions, with best supervision.

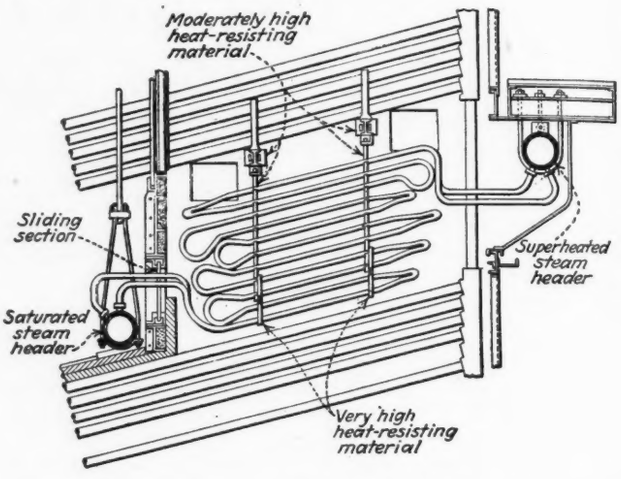


Fig. 2—Example of multiple-loop, single-pass superheater for high pressures and high temperatures

3. Maximum metal temperatures under abnormal and emergency conditions of operation.
4. Minimum temperature at which oxidization starts.

It will be readily seen that the factors named seriously limit the working stress for which a given material may be used. It is generally thought that oxidization starts at a metal temperature between 900 and 1,000 deg. F. Experiments, however, definitely show that at least in the furnace atmosphere destructive oxidization does not occur below 1,200 deg. F.

Calorized Tubing Immune from Oxidation

The manufacturers of calorized tubing indicate that such tubes are immune from oxidation up to a maximum metal temperature of 1,650 deg. F. Thus, calorized tubing appears to offer a relatively inexpensive material for high-temperature superheater construction, of the order of 1,000 deg. F. final steam temperature, at least for the moderately high pressures of 800 to 1,000 lb. per square inch.

The temperature limits herein outlined, of course, require small-diameter carbon tubing with relatively thick walls. The increased temperature stresses, owing to these thick walls, have been taken into account in establishing the design fiber stresses. To reduce the wall thickness required, a fruitful source of investigation would be along the lines of calorizing tubing, which could be produced for twice the cost of carbon steel tubing and have a better high-temperature characteristic. Such tubing is not in sight at the present time.

To obtain such high steam temperatures involves locating the superheater in an even hotter gas zone than is the present practice. When this is done, the supporting problem becomes even more acute than at present. The supporting medium should be so designed that it is in tension under all conditions. When this is done, the

critical part of the boiler is relieved of the weight of the superheater and boiler tube maintenance may be accomplished without in any way disturbing the superheater. Heat-resisting materials of the highest grade must be used to support that part of the superheater in the hottest zone. A low-grade, though very heavy, heat-resisting construction may be used for the remainder of the superheater. A highly successful superheater supporting construction is shown in Fig. 2.

Materials for superheater headers do not present a problem of such magnitude as superheater tubing. Headers can be so located that their temperature cannot be raised to a point higher than that of the contained steam. The headers containing superheated steam should be constructed of the same metal as the superheater tubing containing the highest-temperature steam. Stresses, slightly greater than those given in Fig. 1, for tubing, may be used and the resulting practical factor of safety will be the same as found for the tubing.

Principles of Design

In designing high-pressure and high-temperature superheaters certain fundamental principles must be strictly adhered to if serious difficulty and high maintenance are to be avoided. These principles will be briefly outlined:

1. *High Steam Velocity*: The over-all coefficient of heat transfer is composed of three individual coefficients, namely: Outside, or gas to metal wall; conduction through the tube wall; and from the tube wall to the contained steam.

The coefficient of heat transfer between the inside wall and the contained steam increases as the steam velocity increases up to a certain critical point. At this point the resistance on the steam side is swept away and further increases in velocity have little influence on the coefficient of heat transfer. This critical velocity varies with the size of the tubing, increasing as the tubing diameter increases. Fig. 3 shows the characteristic curve for a given set of conditions. Thus, if the steam velocity

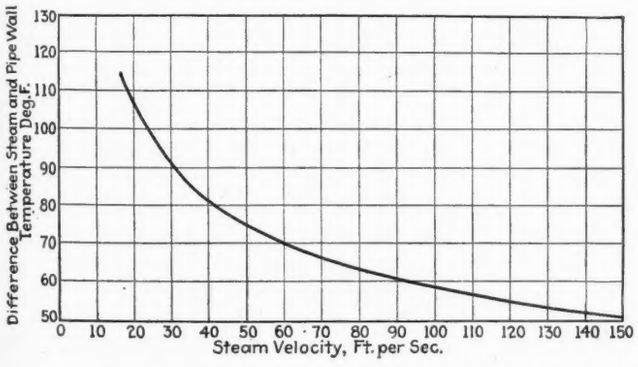


Fig. 3—Temperature difference plotted against velocity

in the superheater is at or near the critical velocity at normal load, the difference in temperature between the tube metal and superheated steam will be the minimum. Therefore, the practical factor of safety of operation will be the maximum for the material used. Thus, the employment of a high steam velocity permits the use of normally available materials at higher temperatures than would otherwise be possible.

2. *Steam Distribution*: Perfect steam distribution at all loads is a highly desirable feature of design under normal pressures and temperatures. With high pressures and temperatures it is vitally necessary. Perfect steam distribution can be obtained only with a multiple-loop, single-pass design of superheater. A typical design is shown in Fig. 2. The inlet to the saturated header must be on the end opposite the outlet to the

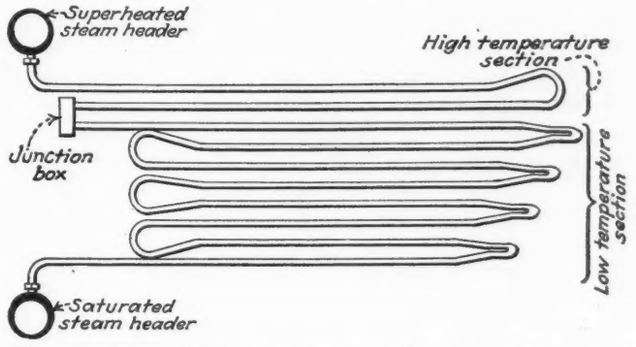


Fig. 4—The high-temperature section is separate and joined through a junction box to the low-temperature section

superheated header. With this construction the length of steam travel is the same for all sections of the superheater; thus each unit must draw its proportionate share of steam and local overheating is therefore avoided.

3. *Use of Several Tube Sizes*: From the foregoing explanation of the effect of velocity on superheater design it will be appreciated that several tube sizes must be available to apply these principles successfully to the innumerable design specifications. This is most vital when the multiple-loop, single-pass design is the standard of construction. The use of the smaller tube sizes, $1\frac{1}{2}$-in. and $1\frac{5}{8}$-in., permits using the low-cost materials at high steam temperatures.

4. *Superheater Unit Design*: The sections comprising the superheater unit should be joined with forge welds. The forged return bend has proved to be a successful solution of this problem. The superiority of forge welding has never been successfully questioned. The radii selected for the 180-deg. bends are also of more importance than generally recognized. Thus, if the bending radius for a certain selected tubing gage and diameter is reduced from 6 in. to 4 in., the metal thickness on the outside of the bend is reduced from 0.124 in. to 0.114 in., the original metal thickness being 0.134 in. Thus large radii must be used, especially when high temperatures are encountered, else it will be necessary to increase materially the tubing thickness in the straight lengths.

5. *Distribution of Superheater Tubes*: It is a well-known fact that the heat transfer in a strictly contraflow design is the maximum obtainable. In superheater design, however, other factors must be taken into consideration. Thus, it is better practice to locate the coolest, or saturated steam, tube in the hottest gas zone. The design then may be strictly uniflow, or the lower section may be uniflow and the upper section contraflow. With such an arrangement the tube-metal temperatures are somewhat equalized. In actual practice the metal temperature in the hottest gas zone, even with the increased heat transmission, is less than the metal temperature of the outgoing tube. With this design a maximum practical factor of safety is obtained.

6. *Division of Superheater into Two Sections*: Whenever the limitations of design permit, the superheater should be divided into high and low-temperature sections. The low-temperature section will, of course, be composed of low-carbon steel tubing and will be used to raise the steam temperature to approximately 800 deg. F. The high-temperature section, composed of special tubing, should be joined to the low-temperature section with a box and through easily detachable joints. Thus, the section of the superheater requiring the greatest maintenance may be readily replaced without disturbing or destroying the largest section of the unit. This construction is shown in Fig. 4.

7. *Header Location*: Whenever a design permits, the superheated steam header, at least, should be located outside of the boiler setting. The practical advantages of this arrangement from the viewpoint of joint maintenance is obvious. This construction is shown in Fig. 2. This feature of construction is even more desirable at high temperatures, since under these conditions of operation the header must be supported at several points throughout its length, and this can always be more readily accomplished outside of the boiler setting. In this position there is also no danger whatever of overheating the headers through the medium of high gas temperatures.

8. *Header Design:* A practical form of high-pressure header is shown in Fig. 5. With this eccentric construction the maximum metal thickness is located in the position most needed. Owing to the use of detachable joints, no handholes are required and a superheater unit may be removed or replaced with great rapidity. Because of the trend toward larger steam generating units it becomes vitally necessary to continue them on the line for the maximum length of time. Rapid superheater maintenance is a contributing factor to the successful accomplishment of this most desirable condition.

9. *Baffle Design:* Where the superheater design is such that one or more of the headers are within the

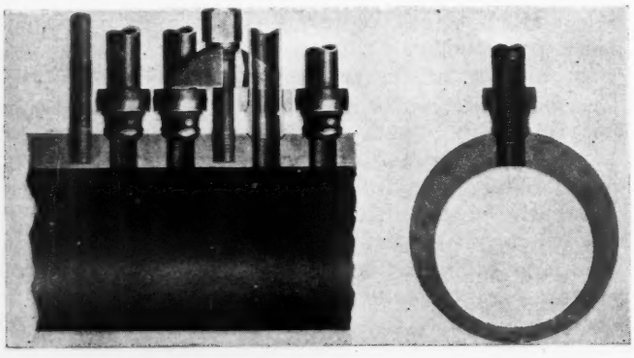


Fig. 5—This high-pressure header is made from a solid billet and bored eccentrically

boiler setting, they should be placed definitely behind a baffle to remove them from danger of exposure to high gas temperatures. This baffle should be of such design as to permit free movement of the boiler with relation to the superheater. A successful baffle design is shown in Fig. 2.

10. *Uniform Superheat:* A fundamental principle of superheater design, where a flat characteristic is desired, is that the majority of the superheating surface must be adjacent to the steam generating surface from which most of the steam is generated. Thus, the superheating surface is exposed to the same fluctuations in gas temperature as the steam-generating surface, which is generating the major portion of the steam.

In addition to this, the effect of steam velocities above the critical velocity, as discussed in the foregoing, is to permit a dropping off in steam temperature at loads above the critical velocity. This is shown in Fig. 6. The peak of the superheater curve is at or near the critical steam velocity. It will be readily appreciated that this shape of curve gives a much smaller variation in steam temperature over a wider range than could be obtained with a design the steam velocity of which at maximum load is at or below the critical velocity. This characteristic gives an additional practical factor of safety for high-temperature superheaters, in that the maximum temperature is strictly limited.

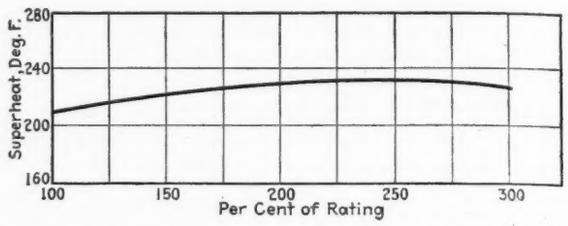


Fig. 6—Curve showing uniformity of superheat over wide load range.

Early Applications of High Pressures

THE employment of high-pressure steam is not so recent as might be imagined, because between 1820 and 1830 Jacob Perkins advocated the use of high-pressure steam from 1,000 to 3,000 lb. per sq.in. in steam engines, and invented a compound uniflow engine to work at 1,400 lb. per sq.in. and 1,000 deg. F. His grandson carried on this work and, in 1859, built a yacht driven by a triple-expansion steam engine designed to work with 600 lb. per sq.in. steam pressure with re-superheating between the cylinders. Again, about 1880 another steam yacht was built by Messrs. R. Steel & Company, of Greenock, to operate with a boiler pressure of 500 lb. per square inch.

After a lapse of nearly 50 years we have a return to the high steam pressures in the steamship "King George V," built for service on the Clyde. This ship was launched in 1925 and has two Yarrow water-tube boilers and turbines built by the Parsons Marine Steam Turbine Company, designed for a steam pressure of 550 lb. per sq.in. superheated to 750 deg. F.

The first power station operating with high-pressure steam was, however, at the Stockholm Exhibition in 1897, with superheated steam at 1,700 lb. per sq.in. Six DeLaval single-stage turbine generators, four of 100 kw. and two of 50 kw., each with its own high-pressure boiler, generated all the electric power required for lighting the Exhibition.

For the next twenty years, however, work on these very high pressures seems to have lapsed and only a very gradual increase in pressure took place until, in about 1917, the North Tees Station of the Newcastle Electric Supply Company was designed to work with a boiler pressure of 475 lb. per sq.in. gage at 700 deg. F. —*From a paper by A. H. Law and J. P. Chittenden before the Institution of Electrical Engineers, London.*

Boiler Operation

AT HIGH PRESSURE

Demands Exact Water Conditioning

By R. E. Hall

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EXACTITUDE in water conditioning is an elemental requisite to successful operation of a modern boiler. Its realization requires that the essential factors comprising it be clearly defined. It is the purpose of this paper to set forth certain of these essential factors and, in particular, to demonstrate their unique relation to water conditioning for high-pressure operation.

TEMPERATURES (DEGREES F.) CORRESPONDING TO VARIOUS ABSOLUTE PRESSURES

Absolute Pressure, Lb. per Sq. In.	Temp. Deg. F.	Absolute Pressure, Lb. per Sq. In.	Temp. Deg. F.
140	353	465	460
165	366	665	498
215	388	1215	568
265	406	1815	622
365	436	3200*	705*

*Critical Point.

The table comprises the temperatures corresponding to certain absolute pressures. In considerations directed to the relationship of boiler metal and water, temperature is the dominant factor. In proof of this it may be noted that solubilities of dissolved substances in the boiler water vary materially with change of temperature, but only minutely with change of pressure when temperature is maintained constant. Inasmuch as scale-forming substances become less soluble as the temperature increases, it is inevitable that the permissible tolerance thereof in the boiler water must be highly restricted at temperatures corresponding to higher operating pressures.

Most important of all, however, temperature is a primary factor in determining type and rate of chemical reactivity between the boiler metal and water. Thus, at ordinary temperatures, iron has but slight tendency to dissolve in pure water and corrosion occurs because different localized concentrations of oxygen accelerate dissolution at one point and retard it at another. With oxygen absent no noticeable corrosion occurs. At higher temperatures, however, and even in the absence of oxygen, direct reaction of iron and water occurs with formation of the magnetic oxides. If conditions are maintained in the boiler water so that the oxide film thus formed integral with the metal is tightly adherent and free from perforations, and so that any incidental rupturing of the film is quickly repaired, then direct reaction of boiler metal and water becomes well-nigh infinitely slow.

Chloride in the boiler water is inimical to impermeability of the oxide film, and the severe corrosive conditions invoked by its presence are too well known to require comment. Consider a patch of relatively thin scale or other impediment to free transfer of heat on one of the tubes of a high-pressure boiler. It is incongruous to suppose that of itself it can cause the thick tube to bulge and thus to fail. One reflects, however, that it represents a localized region of higher temperatures, which abets deterioration of the protective oxide film and accelerates direct reactivity of the steel with contacting water; also that accumulation of iron oxide, resulting from the reaction by further retarding heat transfer, hastens penetration of the metal and final tube failure. Hence, prevention of even this thin patch of scale becomes a tangible asset in assuring more certain and continuous impermeability of the protective oxide film.

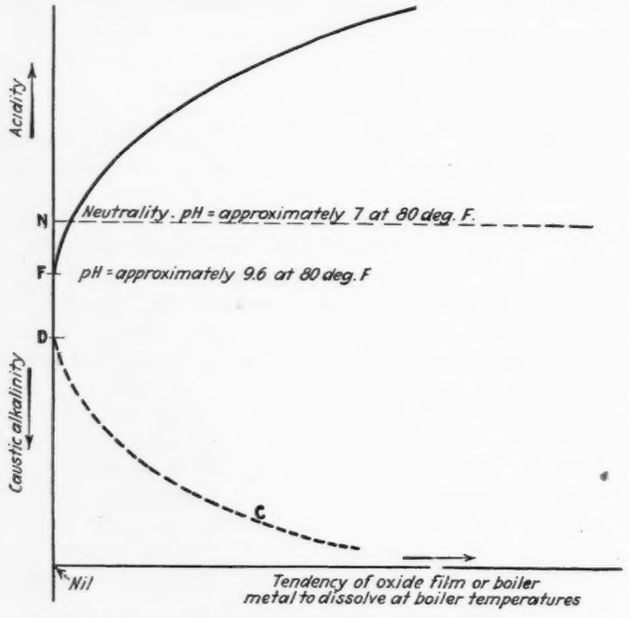
The logical conclusion is as follows: *The function of water conditioning for high-pressure operation is the continuous maintenance of a tightly adherent and fully intact thin protective film that is formed from and integral with the surface of the boiler.* This film defends the metal against its direct reaction with the boiler water and against the attack of any components therein, as chloride or uncombined oxygen in particular. The water conditioning must prevent formation of scale, or the arising of other circumstances that lead to localized or general disruption of the film, and must comprise conditions that assure ready repair of any ruptures that develop.

In thus delegating protection of the boiler metal to the maintenance of integrity and adherency in a thin impermeable film integral with the surfaces it must protect, we are dealing neither with the intangible nor with the unknown. Aluminum sheet, for instance, is protected by a thin film of aluminum oxide and corrodes only at points where rupture of film occurs. Passivity of iron has been proved due to the protective action of a thin film. The steel of a gun barrel is protected by a thin oxide or phosphate film integral with the metal. A strip of iron, heated at one end, acquires a banded color arrangement whose termination occurs at some distance from the point of heating. The colors are due to film

formation causing interference of light, and their arrangement denotes gradual change in the thickness of the film.

With uninterrupted maintenance of an impermeable protective film integral with the boiler surfaces as the central idea of water conditioning, there need be no indefiniteness in defining those conditions in the boiler water that are optimum for the purpose. We will briefly discuss (1) the relation of primary treatment thereto; and (2) the most certain means of insuring against any scale formation; finally, (3) we will discuss the optimum conditions for protective film formation and repair, along with some of the conditions inimical thereto.

No special new problems are involved in the preliminary treatment of water for high-pressure operation. At the intermediate pressures of 400-600 lb. with makeup water varying from a few to 100 per cent, primary treatment by either lime-soda or zeolite in conjunction



Relation of acidity and causticity to protection of the boiler metal

with standard means of oxygen removal and followed by secondary control on the boiler water by phosphate, adequately protects the surfaces and offers no particular difficulties even with a severe water. As the next step to 1,200 lb. or higher is taken, it is advisable that makeup be mainly evaporated water or condensate, since the increased temperature makes more difficult the protection of the surfaces against effects of dissolved components in the water such as chloride. Likewise, at the higher pressure, practically complete elimination of dissolved oxygen should be assured. In brief, the preliminary treatment of water for even the highest operating pressures differs from that for low pressures only in that it must provide for simple and continuous maintenance within much more highly restricted limits of exactly defined conditions in the boiler water.

In spite of all precautions in preliminary treatment, some calcium, sulphate, and usually silicate find their way into the boiler water. Their amount may be small, but their presence casts a reasonable doubt on continued cleanliness of the surfaces and hence on certainty of protective-film impermeability. Insurance against any such doubt lies in maintenance in the boiler water of a negative radical such as phosphate, stable at all operating pressures, which removes the calcium by combining therewith to form insoluble suspended matter, and thus simultaneously renders the sulphate and silicate innocuous. The replacements of phosphate necessary for maintenance of definite conditions bear witness to the amount of calcium coming to the boiler water.

The amount of caustic alkalinity in the boiler water required for most effective results from the phosphate, coincides with that specified below as essential to promote integrity and adherency of the protective film.

The effectiveness of phosphate maintenance has been demonstrated in many plants up to 450 lb. operating pressure, and at 1,200 lb., by John Anderson at the Lakeside Station of the Milwaukee Electric Railway, Light & Power Company.

The accompanying curve presents schematically the stability of iron and any protective oxide film at boiler temperatures in contact with solutions ranging from highly acid to highly caustic.

In defining conditions to maintain in the boiler water, the region of interest is that lying between *F* and *D*. *F* corresponds to a *pH* value of 9.6 in a sample of boiler water cooled to room temperature, and represents the minimum caustic value (hydroxide — 0.7 p.p.m.) at which protection of the boiler metal is assured. *D* represents the caustic concentration at which dissolution of metal or oxide film begins to occur. With increasing temperature the gap between *F* and *D* decreases, a fact that emphasizes the necessity for highly restricted limits of tolerance in boiler waters at the highest operating pressures.

It should be noted that the point *F* lies quite a little on the alkaline side of *N*, the point of strict neutrality in the water.

The point *D* represents a considerable caustic concentration, and hence satisfactory limitation thereof in the boiler water proper, even at the highest operating pressures, presents no difficulties. But chance for evaporation of the boiler water and hence multiplication of the concentration therein in any seam, or at any point at which active circulation is prevented, is a primary reason for limiting caustic alkalinity to the lowest value compatible with integrity and adherency of the protective film. The inimical effect of caustic alkalinity on boiling conditions constitutes a second reason for strict limitation of its concentration.

On the other hand, the lower limit to caustic alkalinity must be higher than that corresponding to *F* in order that phosphate may most effectively remove calcium from the boiler water.

It is our policy, therefore, in those cases in which the boiler water is extremely pure, to make the caustic alkalinity conform as closely as possible to the range 10-25 p.p.m. of hydoxide (OH). If the concentration of dissolved materials in the boiler water is higher, and in particular if chloride is present, a minimum of 25 p.p.m. of hydroxide is advisable, with an upper limit of perhaps 50 or 75 p.p.m. as conditions may require. These limits are a little lower than those we have heretofore suggested, and are made feasible by the exact control methods that have been developed in connection with our water-conditioning work.

The concentrations of hydroxide indicated are far enough on the alkaline side of *F* to insure integrity and adherence of the protective film, and to provide for quick repair of any rupture; they are far enough removed from *D* to discount any direct dissolution of the film in contact with boiler water proper.

Exceptionally high concentration of the boiler water is largely confined to riveted seams and similar areas as points of juncture of tubes with drums or headers, blowout flanges, etc. As evaporation occurs at such points, the problem of maintaining uninterruptedly a protective film that is impermeable and integral with its parent surfaces is confined no longer to the causticity of *FD*, but with increasing concentration gradually drifts into the region *DC*, where the high causticity is not beneficial, but is actually destructive to any oxide film. Likewise, corrosive agents, as chloride, are of similarly higher concentration than in the boiler water proper.

In addition to highly concentrated contacting water, other factors are also effective in complicating the problem of maintaining impermeable protective films in these areas. The metal of butt straps, rivets or the rolled-in ends of tubes is under stress, and subject further to cyclic stress.

As demonstrated by Speller, impermeability of protective film is more difficult of maintenance on steel under stress than when not stressed. Dr. D. J. McAdam, Jr., of the Engineering Experiment Station, Annapolis, Md., has demonstrated that corrosion is accelerated by cyclic stress and that "even slight corrosion simultaneous with fatigue (corrosion-fatigue) may cause failure at nominal stresses far below the ordinary endurance limit."

Failures of the boiler metal occur at these points. Parr and Straub, by collation of the data of numerous failures and by experimentation in the laboratory, have emphasized conjunction of causticity and stress in causing the failure. Williams and Homerberg, of Massachusetts Institute of Technology, paid particular attention to the embrittling effect of hydrogen that might be developed in the seams by interaction of boiler water and metal. McAdam, by demonstrating the importance of *cyclic* stress and corrosion-fatigue to readiness of failure, has shown that all those conditions in the concentrated water that accelerate corrosion must be considered causative of failure.

Whether this type of failure be designated finally as embrittlement or corrosion-fatigue, and whether cracks resulting therefrom be inter- or transcrystalline, there is agreement on one fundamental thing; namely, that inception of ready failure has its origin in chemical attack of the metal by caustic or other corrosive component of the concentrated water. This agreement is extremely important from the viewpoint of water conditioning, since, regardless of differences of opinion on the mechanism of failure, the method of prevention is the same in all cases and resolves itself into stoppage of the chemical attack that leads to inception of failure.

We believe that those substances known to inhibit these failures so function because of their capacity to build and to maintain continuously a protective impermeable film integral with the surfaces in the danger zone; and that differences of effectiveness in different substances are due to the film-forming characteristics of those substances.

For a number of years sulphate has been used as a protective reagent. Recently, Parr and Straub have enlarged the field of possibilities by demonstrating in the laboratory the extreme effectiveness of phosphate. Other materials in their list, as tannates or acetates, are rejected from consideration in this discussion because the instability of organic materials at the temperature of high-pressure operation raises doubt as to their dependability.

The same type of oxide film that functions to protect the boiler metal may possibly be effective in the seams, though the destructive action of concentrated caustic and any corrosive components in the water must certainly render maintenance of its impermeability more difficult. It is possible that the protective action of sodium sulphate may be thus explained, and that the greater concentrations required at higher temperatures are necessary to combat simultaneously increased dissolving action of caustic.

There is less difficulty in picturing the mechanism of protective film formation by phosphate, since the reaction products thereof with the boiler metal are characteristically film-forming.

There are two specific factors in water conditioning directed to prevention of failure in seams or similar areas whether due to embrittlement or to corrosion-fatigue:

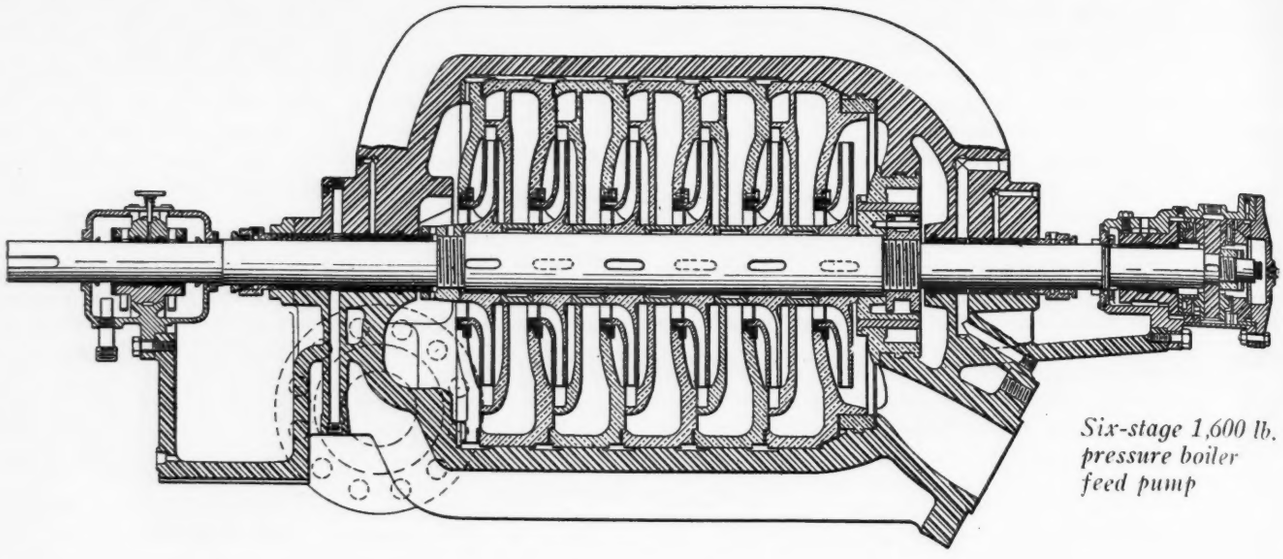
1. Limitation of caustic alkalinity in the boiler water to the lowest concentrations compatible with protection of the boiler surfaces proper, thus limiting causticity in the seam. This may be accomplished by use of phosphate as final conditioning chemical, since it gives independent control of alkalinity and is effective in preventing scale formation at desirable low concentrations of alkalinity in the boiler water. The concentration of chloride in the boiler water should be held as low as possible because of its corrosive properties.

2. Maintenance in the boiler water of dissolved components that shall concentrate therewith in seams or similar areas, and shall prevent failures, whether these result from causticity and embrittlement, or from cyclic stress and corrosion-fatigue.

Sulphate and phosphate are the preferred preventive reagents. Although the former is the more generally used reagent at the present time, the relatively large quantities required, thus inviting troubles from carry-over in the steam, and the possibility of sulphide formation by its reduction, are factors deterrent to its use. Further, its protective action seems due to secondary rather than primary reactions of the sulphate radical. Phosphate, on the other hand, acting in the dual capacity of preventing scale formation and of protecting the metal in seams, offers none of the difficulties mentioned. Furthermore, from the analysis herein given, which ascribes the high protective action of phosphate to its entering directly into impermeable film formation, there is every reason to believe that the superiority of phosphate over sulphate will prove as great in practice as demonstrated in the laboratory by Parr and Straub.

The impermeable protective thin films, maintenance of which the author has defined as the object of water conditioning for high-pressure operation, bear no relation to mechanically applied coatings. The thin films are produced by reaction of the water or a chosen component therein with the metal, and are thus integral with the latter. Their integrity at all times is assured by the maintenance of well-defined conditions in the boiler water, for under these circumstances incidental ruptures of the film are repaired as they form.

In conclusion, the author would emphasize the fact that at the elevated temperatures corresponding to high-pressure operation, chemical reactivity between boiler metal and water is much more intense than at lower temperatures. Hence the limits of tolerance in water conditioning for high pressures must be very restricted, and exactitude becomes a major element in its success.



Six-stage 1,600 lb. pressure boiler feed pump

Meeting Hydraulic and Mechanical Requirements in *Boiler Feed Pump Design*

By O. H. Dorer

Assistant Chief Engineer,
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THE upward movement of pressure in steam power plants has developed numerous problems with every piece of apparatus in the plant, and the feed pump is no exception.

Present experiences point forcibly to a conflict between the mechanical requirements and desirable hydraulic features. This is best explained by stating the working principle of a centrifugal pump. Through the rotation of a wheel containing passages or vanes, water gains pressure and velocity energy in its travel from the center outwardly to the periphery. The nearer to the exact center that this action can start, the greater is the pressure and efficiency of the operation, and, conversely, the greater space used by the shaft passing through the wheel center requiring a greater diameter for the entrance edges of the vanes, the poorer the wheel becomes as a pressure producer. These remarks must be correctly assessed by considering at the same time the quantity of water flowing. In comparison, a boiler feed pump has proportionally many times the shaft size for a given flow of water that any other centrifugal pump has, whereas it should have the least, hydraulically, because higher pressure is wanted.

With demands for high pressure and comparatively smaller quantities the tendency has been to increase the rotative speed as much as possible in order to obtain the highest possible wheel pressure. By multistaging, the total pressures obtained increase directly as the number of stages. With more stages, however, the span of shaft between bearing supports becomes greater, which further adds to necessary shaft diameter in order to obtain an operating speed below the critical speed.

Most of the high-pressure installations have the benefit of normal-pressure systems on their suctions, amounting to 300 to 500 lb. per sq.in., and the pumps themselves have to develop from 900 to 1,100 lb. pressure, giving a discharge of 1,200 to 1,600 pounds.

All high-pressure pumps employ single-suction runners hydraulically unbalanced, with a drum or disk counterbalance at the high-pressure end. This principle of construction permits suction pressure on both inboard, or suction, stuffing box and outboard, or discharge, stuffing box. The summation of the pressure thrusts of all the impellers equates to the counterbalance thrust of the drum and disk having the total pump pressure acting on it and allowing the leak through it to be fairly large.

Returning for a moment to the conflict between mechanical and hydraulic design, the shaft is designed just heavy enough to avoid critical speed, and this causes dead space at the impeller eye amounting to 65 per cent of the eye diameter. This large shafting pre-rotates the water at the impeller entrance and affects unfavorably the pump characteristic, so that paralleling two pumps without hunting is most difficult. In addition, water

passing through return chambers at high velocities cannot be kept in stream-line flow without greatly increasing these disturbances. Fortunately, this system of paralleling has not yet been put into force in high-pressure plants. It receives a good deal of attention in plants operating at pressures up to 500 lb., for which pumps having satisfactory characteristics for paralleling can be made. However, there are limitations in such plants also, particularly where capacities under 400 to 500-g.p.m. per pump are involved. Pumps can be made to parallel at constant speed if automatic throttling of pressure on the discharge is used and where boiler checks are not subject to chattering. Most of the paralleling problems come from attempts to save power and from the use of devices that vary the speed of the units only.

As the size of the high-pressure pump is increased, the

Bronze diffusion ring after 3,550 hours of service

conflict becomes greater and an entirely different system results. Experience indicates that 3,600-r.p.m. pumps of high pressure are limited in capacity to about 900 g.p.m. Some attempts to exceed this figure have resulted in a surging pump action that is severe enough to open up the high-pressure piping. When the pre-rotation in front of the impeller is too great, they will not fill steadily, and when the water reaches the discharge of the impeller, the impact of the surge becomes very great, resulting in a dangerous unit. This is the reason for limiting the capacity of 3,600-r.p.m. high-pressure pumps. It is impossible to obtain passage area with a restricted eye even though high pressure prevails.

With increase in size the spread between stages becomes greater, demanding still larger shafts, which eventually destroys every hydraulic principle of the pump unless the rotative speed is decreased. With 60-cycle motors the speed possibilities are far apart. Below 3,600 r.p.m. the next motor speed is 1,800. This speed demands increased wheel diameter of double the size and weight to realize the same stage pressure. At once the casings become much larger in diameter and the problem of holding them together is more difficult. As an offset, more stages and smaller diameters are used, and it becomes necessary to operate two pumps in series to obtain the full working pressure without using shafts of excessive diameter.

The last installation at Boston Edison employed pumps of 1,600 g.p.m. capacity, and the primary pump was made for about 500 lb. pressure. The high-pressure pump is a series unit employing eleven stages with a motor of 2,600 hp. placed between the two elements and operating at 1,800 r.p.m. for 1,600 lb. ultimate discharge.

There exists a high pressure on the stuffing boxes of the high-pressure unit which is avoided in the single-casing high-speed pump. A considerable amount of development work was unecessary on stuffing-box designs, packings and glands, before this problem was solved. It might be said in passing that a rotary shaft is far more difficult to keep packed than a reciprocating shaft for the same working pressure, and the problem of holding 2,000 lb. pressure on a rotary shaft of fair size and speed is still unsolved.

The headpiece illustrates the construction of the final, or high-pressure, end of the unit. It receives water at 900 lb. pressure and delivers at 1,600 pounds. Note the provisions made for bleeding the water away from the stuffing boxes through a long stepped sleeve close fitting to the shaft. This water is bled to the lower-pressure pump so as to maintain an actual box pressure between 500 and 600 pounds.

The increase in size, using the same general principle as used in smaller units, brought out some defects that were not apparent in the smaller unit. One of these is that bronze impellers do not stand up well on high-pressure and high-temperature water. A special steel was developed, known as Wompco, which appears to give the service expected of bronze. A modification in the grade of this metal was successfully used for impeller sealing rings, which did not give service when made of bronze. It is felt that high temperature is principally responsible for the deterioration of the bronze parts.

The photograph of a bronze diffusion ring after 3,550 hours of service on 350-deg. water, shows clearly the erosion of metal, which is just as severe at the section of chamber in back of the impeller as in the vane passages. Note particularly that the tips of vanes are intact, showing that their form is substantially correct for the water flow.

Other experiences that developed concern hydraulic thrusts. As stated elsewhere, there are definite unbalanced thrusts at the impellers, which are counterbalanced by a revolving piston having full discharge pressure acting on it and having an operating clearance as small as possible, to minimize leakage. However, with the increase in stage pressure it is found that the pressures along the outside faces of the impeller disks are not alike and are subject to variation with changes of flow through the impellers. These unbalanced pressures acting on the full surface create a total thrust force amounting to almost 600 lb. per impeller. It is further found that these forces increase in magnitude with each succeeding stage. This requires additional counterbalance over and above the figured counterbalance, and an outside heavy mechanical thrust bearing is used.

Other features of these high-pressure pumps that seem unusual are the omission of shaft sleeves, it being impossible to prevent leakage between sleeve and shaft, and the use of stainless-steel or chromium-plated shafts.

BOILER AUXILIARIES

Keep Pace with High-Pressure Trend

THE old saying that a chain is no stronger than its weakest link can be applied to the boiler house. No matter how well the boiler may perform, safe and successful operation cannot be expected unless safety valves, feed-water regulators, water columns, and blow-down valves all function properly. Thus, the design and construction of these auxiliaries for high pressure have had to keep pace with the advances made in high-pressure boiler design.

Many of the difficulties envisioned in the first high-pressure design did not materialize in practice, while other unexpected troubles developed. Most of these have been overcome so that these boiler auxiliaries now contribute their share to the successful operation of high-pressure plants.

The question of feeding high-pressure boilers with feed-water level regulators have presented some difficulties. It has been found by some that instead of using one large valve, it is better to use several smaller ones in parallel, as these work more easily than one large one.

Unusual high excess feed pressure has been the cause of a large part of the boiler-feed problem. The normal excess pressure of 25 lb. has, in some instances, jumped to as high as 500 pounds.

With these high excess pressures the velocity of the water flowing through the valve has had a tendency to unbalance it, thus preventing the close regulation secured at low pressure. Constant differential regulators for use in series with the level regulator have been designed to overcome this problem. One manufacturer, however, developed a hydraulic relay introduced between the mechanism actuated by the water level and the regulating valve. This leverage provided sufficient force to operate the valve when unbalanced by abnormally high excess pressure.

Changes in the mechanical design, in addition to increasing general dimensions, have had to be made to the various types of regulating valves in order to meet the needs of 1,000-lb. service. Valve-stem stuffing boxes and packing have had to be improved to prevent leakage, and in one type of valve, the additional stuffing-box friction made necessary an outboard thrust ball bearing. Another manufacturer has added an extension to the valve disk which throttles the water while the main disk is lifting clear of its seat, thus preventing wiredrawing and erosion of the main valve seat. The extension to the valve disk made necessary an increase of two inches in the lift of the valve.

It has been found that, so far as the boiler operating characteristics are concerned, feed-water regulation is even more satisfactory on boilers operating at pressures of over 1,000 lb. than on those operating at lower pressures. The density of the steam and water is more nearly equal, and furthermore, the feed water is being supplied to these boilers at a temperature more nearly corresponding to the steam temperature, so that the introduction of feed water does not disturb the operation of the boiler.

Steam pressures of 1,000 lb. have not occasioned radical difficulties in safety-valve construction other than the fact that the number of sufficiently large boilers carrying this pressure, and available for use in testing and developing such valves, is very limited.

Steam erosion of the valve seats at this pressure is not seriously greater than 400 or 600 lb., the total superheat steam temperature carried being little, if any, higher. However, it has been found that 4 per cent blowdown is still desirable for best operating practice, and this at 1,000 lb. pressure, amounts to 40 lb. blowdown. To avoid this loss of pressure and steam, means are generally taken to reduce the frequency of safety-valve blowing. The use of small pilot or warning valves or whistles, various combustion controls, or an auxiliary relief valve, which is electrically controlled to operate with less than 10 per cent blowdown, all help to reduce the frequency of the spring safety valve blowing.

As in the case of many other fittings the safety-valve seat sizes are reduced in general from the 4-in. average size at lower pressure to 2-in. or less at 1,000 lb. This does not, of course, involve more valves per boilers, since the greater density of the steam offsets the decrease in valve size, and results in the same number of pounds per hour capacity.

The development in high-pressure safety valves in the last few years has been of somewhat more interest in connection with high-pressure cracking plants for oil than for steam boilers. The ordinary working temperature for oil in new installations is in excess of 900 deg., and generally averages about one thousand. This high temperature, coupled with pressures which include 1,500 lb., has required a change in the design of safety valves.

First, all working parts must be symmetrical so that they will not distort under temperature changes and, second, the valves must be so designed that seats and disks are free from the unavoidable stresses that distort the bodies, these stresses being in the nature of unequal expansions of the irregular bodies, pressure and piping strains. The high temperature necessitates special designs to protect springs from heat so that a constant popping pressure may be maintained. Similarly, the metals used in these valves, particularly for seating surfaces, require attention, and it is now generally accepted practice that forged seats and disks of alloy steels are used. The necessity for bottle-type safety valves in this oil service can be appreciated when it is considered that the fluid at this temperature only needs the proper amount of air to become quite inflammable. The conditions in oil service have been so exacting that the same valves, when used on steam, give good performance.

As pressures increased, water-column gage glasses began giving trouble. At 400 to 500 lb. the ordinary tubular glass, it was found, had a life of from seven to fourteen days. Such excessive maintenance demanded the development of new types of gage glasses, and for these and higher pressures the flat-type glass is the only type now in successful use. It is constructed of

two flat glasses clamped between properly machined steel housings. Mica is placed between the water and the glass surface and protects the glass from the dissolving or erosive action of the water. The mica, however, becomes clouded, thus interfering with proper observation of the boiler water level, and for this reason it must occasionally be replaced. For pressures of 1,000 lb. these glasses are limited to lengths of about 7 in., thus requiring three sections in series to give a proper picture of the water level in the boiler drum. This limitation in length is due largely to the difference in expansion between glass and steel. The installation of flat glasses for these pressures requires great care and generally cannot be done by the average mechanic.

The water columns used for these high pressures are made of heavy-wall steel tubing and all connections except gage plugs are flanged. Bronze for valves and gage-glass housings has been discarded, and steel has taken its place. The water-column shutoff cocks or valves have required some special attention. These have had to be designed with greater care so that they do not freeze at the higher operating temperatures.

It has been found easy to determine whether there is water or steam coming out of the gage cocks. This is a question that has been discussed and on which there has been some doubt, but it appears that at these high pressures it is even easier to tell the difference between steam and water than at lower pressures.

Water columns with high- and low-water alarm mechanisms have not been furnished for pressures above 750 lb. While manufacturers feel they can provide floats that will stand these pressures, they are doubtful regarding the life of the needle valve that operates the alarm whistle. In spite of this several manufacturers are ready to furnish alarm mechanism with their water columns.

At present it is the practice to operate the boiler blowdown valves only after the pressure has dropped considerably below 1,000 lb. This has been made possible largely because these high-pressure plants used distilled water for makeup. As a result no difficult problems have come up in connection with boiler blowdown valves, and little is known about the probable life of the valve seats should it become necessary to abandon the present practice.

No radical changes have been made in the design of blowdown valves to meet these operating conditions. Greater attention has had to be paid to valve-stem packing and in the piston-type valve a greater overlap between the port openings is required. In some cases a reduction gear has been applied to the valve stem in order to make it easier to operate the valve against the heavy thrust produced by the higher pressures.

A new valve designed particularly for high-pressure service has recently been made available. Its single forged steel body houses both a plug cock and a double-seated globe valve arranged so that one disk is drawn completely out of the water flow before the other starts to open.

When designing a high-pressure boiler, the boiler surface is kept as small as possible and the heating surface necessary to reduce the flue gas to an economic temperature is distributed among the less expensive air preheater and economizer surfaces. This and high ratings have resulted in a greater draft loss through the combined boiler, economizer and air preheater and a consequent increase in the size or speed of the induced-draft fans.

Some High-Pressure and High-Temperature Plants Abroad

ABROAD, particularly on the Continent, there are a number of high-pressure installations. Many of these, however, are smaller units and there are few large central stations that employ extremely high pressures. On the other hand, many plants are running under steam temperatures considerably in excess of practice in the United States. Although the units are in general, smaller reports indicates that no very serious troubles have been encountered.

Heading the list is the installation of two Benson boilers at the Siemens-Schuckert Works, Berlin, where steam is generated at 3,200 lb. The steam is reduced to 2,400 lb. and then heated to 840 deg. F.; it then passes to an Escher Wyss two-barrel turbine for which the steam is used for process.

An Atmos boiler installed in a factory at Stockholm generates steam at 1,500 lb., 760 deg. F. This unit is relatively small.

At the Vienna Locomotive Manufacturing Company, (Austria) there is in operation a Loeffler boiler which supplies steam to an engine at over 1,500 lb. and between 800 and 900 deg. F. Also, at the Witkowitz Coal Mining Company, in Czecho-Slovakia the boilers are reported to furnish steam to a Rateau turbine at around 1,700 lb. and 900 deg. F.

Sulzer Bros., at their works in Winterthur, have in operation a small boiler (8,000 lb. per hour) at 1,500 lb. pressure. Another small boiler of the Hartman type is operating at Ludwigshafen, Germany, at slightly over 1,100 lb. There are also several Schmidt boilers operating in Germany at around 880 lb. and 800 deg. F.

In Egypt also is to be found a high-pressure installation. This is at the works of the Egyptian Salt & Soda Company, where a B. & W. type boiler is operating at over 1,100 degrees.

A plant of unusual interest in England is that now under construction for the Synthetic Ammonia and Nitrates, Ltd., at Billingham, which will contain six units, each of 270,000 lb. of steam per hour, at 750 lb. and 850 degrees.

The foregoing are practically all industrial plants. Among the larger central stations the pressures are more conservative. The highest pressure is that employed in the addition to the Valley Road Station at Bradford, England, where steam is supplied to a 2,500-kw. high-pressure turbine at approximately 1,000 lb. and 800 deg. It exhausts at 210 lb. to a 25,000-kw. low-pressure unit.

Another British plant to employ high temperature and a relatively high pressure is the Battersea station, which will employ 875 deg. F. and 645 lb. pressure.

Next comes the well-known Langerbrugge station of the Centrales Electriques des Flandres, in Belgium, which has been in successful operation for several years at 800 lb. and 840 deg. F. There is a later station in Amsterdam, Holland, which operates at 620 pounds.

The Klingenberg Station in Berlin contains sixteen boilers operating at 515 lb., 780 deg. F., or 750 deg. at the turbine throttle. A like pressure is employed at the remodelled Charlottenburg station.

In South America an outstanding station is that recently completed for the Compania Hispano at Buenos Ayres, which operates with steam at 600 lb. 780 deg., supplying two 35,000-kw. turbines.

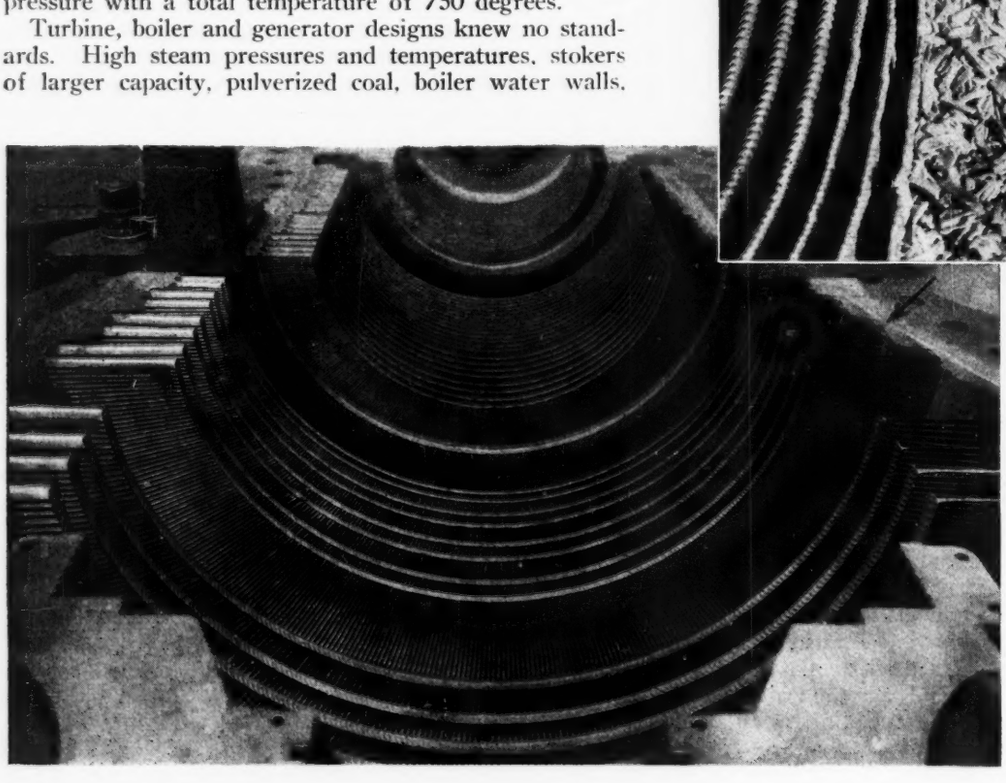
Turbines for High Pressures a

By Bert Houghton
Operating Superintendent, Brooklyn Edison Company

Some of the major turbine difficulties that have appeared in operation have been in connection with spindle designs, steam chests, pipe anchors, case distortion, vibration, blade material and the materials used for casings and blade rings. The primary cause of several of these difficulties lies in the uneven distribution of steam temperatures

IN THE early '90s the operating companies were calling for larger generating units and the electrical and mechanical engineers were making studies to meet these demands. In 1900 we were guessing whether the turbine would ever pass the reciprocating engine in size and efficiency. By 1903 turbines of 5,000-kw. capacity were being built, using steam of 200 lb. pressure with 125 deg. superheat at the throttle. Rapid progress was made in both turbines and generators up to 1914. During the war period, from 1914 and up to 1918, development was slowed up and the sizes held to a maximum of 30,000 to 35,000 kw. The operating people at this time took the opportunity to call on the manufacturer to build larger and more efficient units. The turbine manufacturers had practically reached their limit with 200 lb. steam pressure and began to call for higher pressures. This forced the boiler manufacturers to build boilers for higher pressure, superheat and greater capacity. The boiler people hesitated to build boilers carrying over from 400 to 600 lb. steam pressure with a total temperature of more than 650 deg. The rapid growth of the electrical industry did not allow for conservatism in design, however, and a great revolution in power-plant equipment set in The boiler manufacturers soon found means of fabricating material that would stand higher temperatures and pressures. This enabled them to design boilers that would carry safely 1,600 lb. pressure with a total temperature of 750 degrees.

Turbine, boiler and generator designs knew no standards. High steam pressures and temperatures, stokers of larger capacity, pulverized coal, boiler water walls,

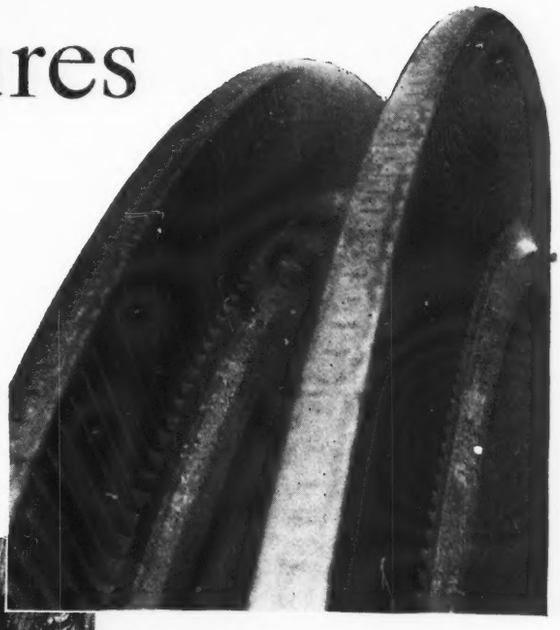


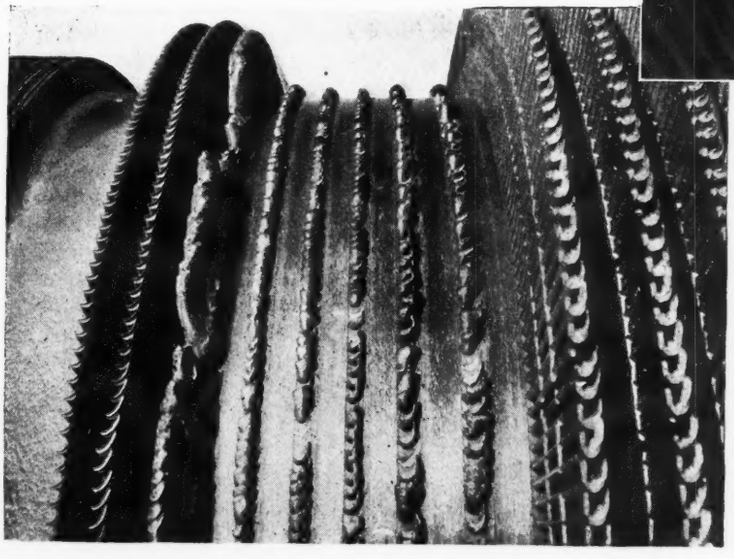
Above — Large turbine blade ring showing damage to intermediate stationary blades after failure of stainless steel blading

Left — Turbine case and stationary blading. Arrow indicates point where cast-iron case has disintegrated

s and Temperatures

From an Operating Point of View





Above — Blading of large turbine showing corrosion-erosion

Left—Large turbine spindle showing intermediate section after failure of stainless steel blades

improved economizers and air preheaters all came in tidal-wave fashion. Manufacturers were unable to complete the development work in their shops, so that many improvements in material and design had to be made after the units were assembled on the customers' premises. In this way a great deal of the expense and inconvenience of development work was shared by the operating companies. The expense, however, was only one part of the operators' contribution to developing the equipment. Both from their experience with the equipment and also because of their interest and eagerness to be of service to the industry, the operating men have made many valuable suggestions regarding the design and materials used in building the larger turbines and boiler equipment. The degree of development and progress which has been attained could never have been reached in so short a time without the assistance of the operating men.

Case distortion, one of the problems imposed by modern operating conditions, is due directly to uneven expansion. That this distortion takes place, there is no question; the character and extent to which it may go is shown in the accompanying curves, the data for which were taken on a large modern turbine Case distortion may lead to serious trouble, as it interferes with turbine alignment and is one cause of rubbing as well as a cause of some vibration. Another serious trouble, which has been more or less overcome by the suggestion of operating people, is the warping of the case out of line due to the steam-chest anchorage and attachment to the casing. The problem involved is in the proper anchoring of the steam chest and piping in its relation to the turbine, so that the turbine itself is free to move at will. The erratic movement of the turbine's casing may be compared in some measure with that of a snake's body. Further reference to the diagram showing the extent and location of the casing movement will bring this out more forcibly.

Of these several causes of trouble the failure of the rotor in its entirety to give proper service has been the most difficult to overcome and is the most serious. The difficulties with the spindle and disks have been mostly overcome, but the blades are still far from satisfactory. The history of turbine development has shown that the temperature limitations of material have been the principal limitations in deriving the highest theoretical economy from the heat cycle. Temperature itself, however, is not the only limitation when it comes to the larger turbines that are in use today. The blading on the intermediate and low-pressure stages is not subjected to high temperatures, but it is subjected to extreme mechanical stresses which must be withstood by materials that also can withstand water erosion, corrosion and vibration.

While this article deals chiefly with high-pressure, high-temperature turbine elements, a general historical background of the development of turbine blading itself is necessary in order to obtain a proper perspective of present-day uses, particularly as we have not as yet reached the ultimate desirable qualities for turbine blading. By ultimate desirable qualities I mean turbine-

blade material that will stand up under operating conditions at least half as long as the expected life of the machine itself, which definition is short of the ideal material. In addition to the discussion of turbine blading, the material used in casings and for other stationary parts exposed to the action of steam must be discussed.

The thermodynamic trend of turbine development has

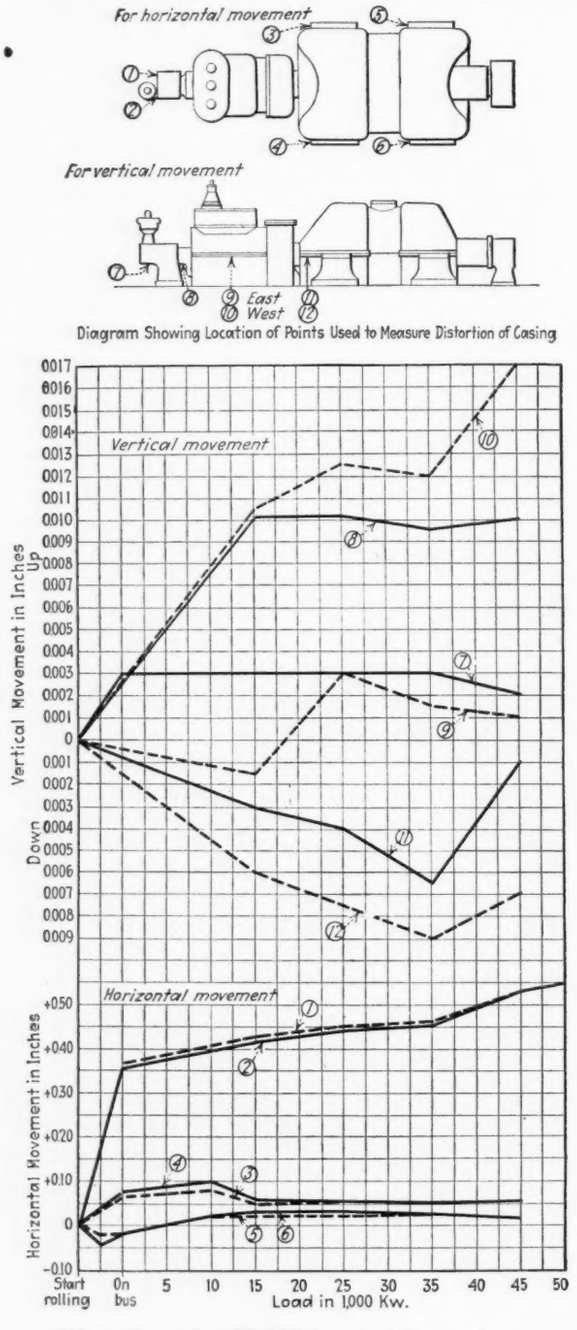


Diagram Showing Location of Points Used to Measure Distortion of Casing

Distortion of a 50,000-kw. turbine casing

been attended by higher steam temperatures and by greater blade stresses. The material has correspondingly had to follow these trends. In the first turbines blade material was not considered particularly important because steam temperatures and stresses were low. Ordinary steel or brass blades were in common use. However, with increasing steam pressure and temperature the early blade materials were found to be entirely unsatisfactory. As this thermodynamic development continued, a whole list of blading alloys were tried and used for a period, but finally were discarded.

Such metals as cartridge brass, aluminum bronze, nickel brass, delta metal, phosphor bronze, manganese bronze, nickel steel, stainless steel, nickel stainless steel, Monel metal and nickel have been used for blades. Only a few of these will withstand the hard usage required in the present-day turbine. Failure of turbine blades is not at all uncommon, and it has taxed the manufacturers to the utmost to produce suitable alloys. There are so many destructive factors, particularly in connection with the larger turbines, that it is usually found that an alloy that is satisfactory in several particulars is deficient in at least one.

Materials for High and for Low Working Stresses

Many of the materials listed here are obviously unsuited for present-day temperatures and working stresses. The alloys being employed may be divided into two major classes, those used for high working stresses and those used for low working stresses. An intermediate field might also be added. In the low-stress field manganese copper and manganese bronze are the leaders. Monel metal and nickel are used for intermediate stresses, while nickel steel and a variety of stainless steels are used for the highest stresses. Phosphor bronze was used in competition with manganese copper until recent years, and many phosphor bronze blades are still in service. I have said that turbine blade alloys which are satisfactory in many respects are deficient in at least one It is unfortunately as true today as it has been all during the period of turbine development. If we consider each of these materials purely from an operating point of view, we find many instances of failure in manganese copper, manganese bronze, nickel steel and in some stainless steels. Again, from the manufacturing viewpoint it is well known that some of the better alloys are extremely difficult to process.

From the operating man's point of view the situation is indeed a serious one. There is no one turbine blade material on which we may depend. We may expect a turbine outage on account of blade trouble almost any time. For protection the operating companies must tie up a great deal of capital in spare capacity in order to guarantee continuous service. Another rather discouraging feature of the situation is, that after serious turbine-blade failures, we are still reblading the turbine with materials that we have every reason to expect will fail again from one cause or another. Nickel blading, when properly fabricated, is remarkably serviceable in the turbine. We may, therefore, expect a certain measure of relief due to the fact that defective nickel blade stock is becoming a thing of the past and that the processes of working nickel and making blades from it are becoming more highly developed.

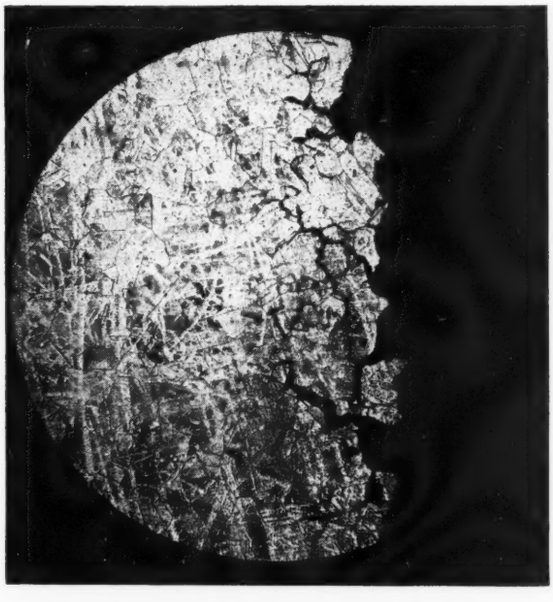
Causes of Blade Failures

The causes of turbine blade failure may be divided into six classes: (a) Vibration; (b) corrosion-erosion; (c) defective material; (d) attack by extraneous gases; (e) embrittlement; (f) rubbing due to blade ring and case distortion.

Turbine manufacturers have appreciated the destructive results of vibration for many years, and a great deal of research work has been done by them in an effort to understand this very complex problem. The determination of critical speed of the disks and the combined

structure and the arrangement of the parts so that the turbine speed will fall between two resonant peaks, have no doubt materially reduced the number of blade failures and practically eliminated the disk failures. Another most important angle from which the vibration type of failure has been attacked, is to produce alloys that will most effectively withstand vibration and consequent failure. However, from an operating point of view, while at the same time appreciating the manufacturers' efforts, we thoroughly realize that this most important problem of turbine blading has not been solved. This is evident from the number of failures which with great frequency take place all over the country today. In a recent meeting of operating men it developed that the failures of vibration types are extremely common, most of them taking place in stainless steel blades. This failure was not particularly prevalent with any one make of turbine, but was confined to one grade of stainless steel. This has been rather unfortunate because in most respects the particular type of stainless steel referred to has stood up remarkably well, in its resistance to corrosion, its high tensile strength, its freedom from chemical attack and its ability to be silver-soldered, which makes it one of the most satisfactory blade materials. However, when it does fail, it causes a rather serious wreck, usually carrying several rows of blades with it which demands a major repair job.

Vibration failures in the manganese copper type of alloy have also been apparent in recent years, but the infrequent throwing of a blade in this very soft material



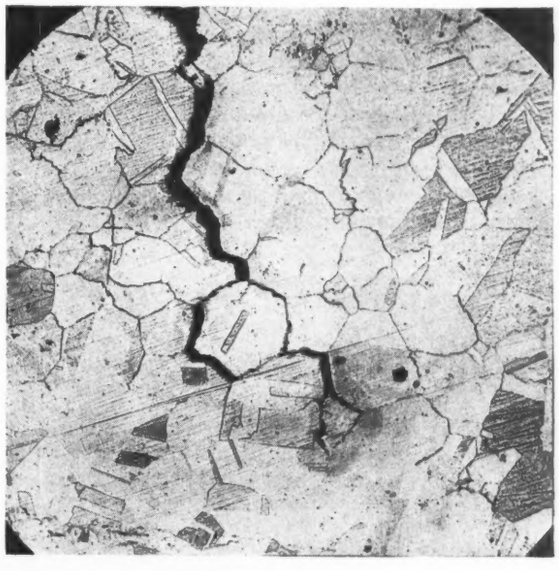
Photomicrograph of manganese-copper turbine blade, showing intercrystalline weakness due to dissociated steam

has not caused any serious damage. Vibration failures in a nickel chromium class of stainless steel have not been at all frequent; however, there has been some difficulty in silver soldering this material. It is being used with success in some cases, and this defect may eventually be overcome in all cases. Among the high working stress materials nickel steel would seem to be the least affected by vibration.

Corrosion-erosion types of failure have been serious, particularly in the ferrous type of alloy. This, primarily caused by oxidation, is one of the chief disadvantages of nickel steel. The failure may take place while the machine is at rest owing to leakage of the throttle or to the infiltration of air through improperly sealed glands. It is also caused by the impurities or wet steam carried over from the boilers. Whatever the case, it makes the use of nickel steel rather unsatisfactory.

Hydrogen in Steam

The attack by extraneous gases in the steam has been a subject that has caused some controversy in the last few years. Some investigators have claimed that the



Photomicrograph of phosphor-bronze turbine blade showing intercrystalline weakness due to dissociated steam

steam did not dissociate to any appreciable extent. Others have argued that in laboratory tests only a few metals were subject to this attack, that such metals as phosphor bronze and manganese copper would not deteriorate in the presence of hydrogen. However, one of the most serious cases that have come to our attention has been the *embrittlement of phosphor-bronze blades.* We have also seen other cases in which the manganese copper has become embrittled. We feel that while the results of laboratory investigation are important and interesting, the results of actual service in the turbine are the data on which we must base our conclusions.

Vibration May Cause Embrittlement

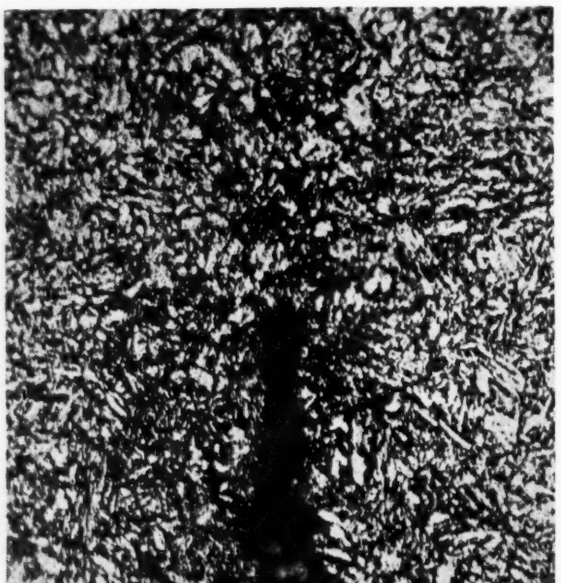
The embrittlement of blades at the lashing wires may, of course, be associated with vibration. This type of failure, however, is not always caused by vibration. There is, in the process of silver soldering the lashing wire, every possibility for heat treatment of the blade material. There is also the possibility of penetration of the metal by the solder. Many materials are subject to this action. There is the possibility of an imperfect weld, which, when broken, causes an undue vibration at this point, embrittling the blade.

Defective material is not so common as it was formerly, but microscopic cracks, nicks and dents in the surface of the blade, which eventually will result in the blade failure, are still found. Manufacturers are giving considerable attention to the prevention of this type of

failure, and they are using every endeavor to produce a smooth blade without these defects.

I have mentioned anchoring the steam chest and methods of attaching it to the turbine casing. The correct solution to this problem has necessitated a long pull on the part of operating men. A rigidly attached steam chest to the side or bottom of a turbine is the great handicap to the free uniform movement of the turbine case, when it is contracting or expanding. Many cases of blade rubbing and vibration have been directly traced to this cause.

One phase of turbine development in which I personally have been extremely interested is cast-iron versus steel casings. The earlier turbines used cast iron exclusively for the cases. After these had been in service a number of years, considerable difficulty was experienced owing to the fact that the cases grew longer than their original dimensions and caused rubbing. Turbine manufacturers were finally persuaded, to use steel casings for the high-pressure ends of all turbines. The use of steel has now extended to the intermediate section. As a result the former difficulty, due to growth of the



Photomicrograph of a stainless-steel blade showing microstructure and typical crack caused by vibration

cast-iron casing and the unequal coefficient of expansion between the steel spindle and cast-iron casing, has been eliminated. The cause of this growth of cast-iron casing I have always attributed to hydrogen or other extraneous gases in the steam. It has been well proved that hydrogen causes disintegration of cast iron, often causing growth, and finally failure. This has been particularly evident in cast-iron steam fittings. There is no reason to believe that this same action does not take place in cast-iron blade rings in the turbines, causing distortion in the alignment of the blade, which finally results in blade rub.

When it is considered that certain metal failures are caused, not by the steam temperature, but by the permeation of hydrogen present in the steam, the reason for failure of turbine casings is more clearly understood.

In spite of the present trend away from the use of cast iron for turbine casings, particularly on the high-pressure end, there is still room for improvement and the time is at hand when the enlightened turbine manufacturer will relegate cast iron as casing and blade-ring material into the discard.

One feature of turbines which should be stressed is the subject of governors. This is particularly important nowadays with our large interconnections and the requirements of our customers for accurate frequency in using electric clocks. Many articles have been written on the design and characteristics of a variety of types, as well as applications and sensitivity of turbine governors. The one common attribute of all governors, which the studies point out, is that a change of speed is required before the governor will function. On the high-pressure turbine the governor not only is required to maintain accurate revolutions per minute, but also must be sensitive to the load requirement of the turbine with regard to the heat balance of the station. It may be said that the governing of the machine itself has been worked out in a highly satisfactory manner. The point that is not so well understood, however, is that the system frequency and individual turbine speed is to an appreciable extent dependent upon uniform steam conditions from the boiler room. Where a fluctuating frequency is permissible, say a half-cycle either side of normal, and this fluctuation is gradual, the boiler situation is not particularly important, but at present, where a minimum magnitude and period of fluctuation of frequency are demanded, the boiler pressure becomes an extremely important item. I have stressed this before in other papers, but wish to reiterate it at this time.

Constant Steam Pressure Important

In the operation of generating stations, particularly in interconnected systems, the steam pressure is usually passed over with a suggestion that the governor action is sufficient to compensate for changes in steam pressure, even though the pressure changes are comparatively large. This is, however, only partly true owing to the difference in the behavior of governors on machines operating off the same steam headers. Some operating companies maintain that it is mostly a matter of discipline to maintain good regulation. Even if this were true, constant steam pressure would lessen the burden of the man in the control room and save the excess wear and tear on the governing mechanisms, contacts and so forth. It is the steam source we must regulate. The turbine governor is not sensitive enough to act and maintain close regulation with other units in the same station, although the units are built in the same shop under the same specification. I have been of the opinion for several years that automatic governors should be placed on the boilers to maintain the required steam pressure and that they are just as important as the automatic governors on the turbines.

In concluding this résumé of high-pressure turbine construction and the development that led to the present possibilities, I wish to point out that, while great strides have been made and while we have the equipment with which the electrical supply may be economically produced, there is still a great field for concentrated effort by both the operating companies and the manufacturers to produce a turbine that may be viewed with a greater feeling of assured continuity of service.

The Steam Engine as a High Pressure Industrial Prime Mover

By John F. Ferguson
Ferguson Engineers, Chicago

One is accustomed *to think of turbines in connection with the high-pressure plant, which, of course, is true for the central station. In the industrial plant, however, the reciprocating engine has been used successfully in Europe with steam at 1,500 lb. pressure at the Vienna Locomotive Manufacturing Company's plant. Also the highest-pressure plant in the United States will employ reciprocating engines with steam at 1,800 lb. pressure. The high efficiency of the engine in the high-pressure range indicates a field for this type of prime mover, and Mr. Ferguson points out some of its advantages.*

FOR industrial plant service calling for high initial pressure and high back pressure the reciprocating engine has many advantages. The higher these pressures at either end of the steam cycle, the greater the advantage of the steam engine over the turbine. However, the engine must be designed to meet these new conditions.

In prime movers for the industrial field the advance has been slow. While numerous types have been developed and improvements on each of these types all represent experience and progress, it may be well first to review briefly some of the history of steam prime movers leading up to present conditions.

At the beginning of the Twentieth Century the reciprocating steam engine of the compound type predominated in the higher-class installations. The steam-turbine era was scarcely under way. Since then great strides have been made in the steam turbine; it has displaced almost entirely the engine unit in public utility and other condensing service and made great inroads in the back-pressure service of industrial plants.

In the early years of the turbine growth, the low-pressure turbine gained a place, particularly in connection with existing engine plants, by converting non-condensing into condensing plants and increasing the capacity and the flexibility of operation. When properly applied such an installation represented an ideal combination for high efficiency, but this arrangement was greatly abused in application and operation, so that the great advantages were seldom obtained.

With the introduction of the non-condensing or back-pressure turbine on auxiliary units, many problems pertaining to steam power drive for high-speed units were simplified. As a power unit this type of prime mover has been most satisfactory where the steam demand for process was large and the power demand low, or where no real efficiency in the prime mover was required.

Then came the bleeder or extraction type of turbine with a combination of back pressure and condensing operation. This "cure-all machine," as one turbine manufacturer called it, was to fit almost any condition of back pressure and condensing service, but rarely was best suited for the specific conditions in the plant in which it was installed.

In the past higher efficiencies have been sought in the low-pressure stages or at the vacuum end of the cycle, but today the endeavor is to improve efficiency by the use of higher initial pressures and the elimination of the lower stages of expansion to avoid condenser losses.

With the introduction of the higher steam pressures will be seen a new line of prime movers of the reciprocating type, particularly in the industrial power field. Here back-pressure operation will prevail and condensing operation will be negligible.

With the extremely high steam pressures already in use or under consideration, we have gone beyond the reach of some of the standard types of engines, have passed the best field for the turbine and entered the best field for the engine unit of suitable design.

In the engine there are characteristics and features of construction that make it ideal for high working pressures and less favorable for low-pressure service. The characteristics of the turbine point to best performance on low working pressures and less favorable results from extreme pressures.

With engine designs available today, some of the former advantages of the turbine, such as less space occupied, have been reduced to a small margin. As to reliability of service, long continuous operation without stop or adjustment, long useful life of the unit, the engine seems to hold the advantage by a considerable margin.

Initial investment for the prime mover, only, of a given capacity, may be less for the turbine than for the engine unit, but if consideration is given to the total investment for a complete plant, with boilers, prime movers, etc., included, having steaming capacity and pressure sufficient to meet a given load condition, then the total investment will be decidedly less for the engine installation, as equivalent results may be obtained at considerably lower pressures.

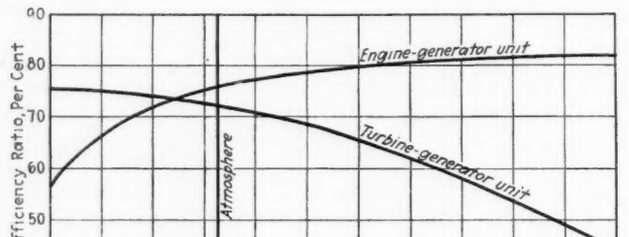
With the reciprocating unit it is possible to go to high pressures without any particular trouble. In this country compressors are developing pressures of 3,000 and 4,500 lb. and in Great Britain and Europe much higher pres-

sures. There is no reason why the cycle cannot be reversed and expansion carried downward from pressures of 3,200 lb. The pressures referred to were obtained with air and gas mediums that are more difficult to work with than steam.

As to steam temperatures suitable for the reciprocating engine, there are no reasons why it should not be possible to go to, or even beyond, the maximum of present-day practice with turbines, and certainly to the economical limit of the entire steam plant in any case.

In the accompanying diagram is indicated the relative efficiency ratios of the engine and turbine, of the higher class in each case, that may be expected with favorable load conditions under any steam pressure range. Combined efficiencies have been shown because it is the combined results from the prime mover that govern. These curves indicate the combined efficiency, thermal, mechanical and electrical, at stage pressures throughout the range of expansion, and are based upon the following approximate mechanical and electrical efficiencies of the units:

Mechanical efficiency of engine unit, per cent	93 to 94
Mechanical efficiency of turbine unit, per cent	97.5
Electrical efficiency of engine generator, per cent	95
Electrical efficiency of turbine generator, per cent	95.5



Combined efficiency ratios for engine and turbine units at various pressures

It must not be taken that any point on the curve represents the efficiency in expansion from corresponding pressures to the final or extreme release pressures, but refers to the efficiency within a narrow range of expansion near the specific point. For instance, the average efficiency ratio of the engine-generator unit expanding from 200 to 50 lb. pressure approximates 80 per cent.

The curves are drawn to cover fair-sized industrial power units of around 2,000 kw. capacity using about 40,000 lb. of steam per hour. With larger units the efficiency curves would be slightly higher in each case. The efficiency of the turbine unit would increase slightly faster with an equal increase in capacity than that of the engine, while with smaller units the turbine efficiency would drop much faster than that of the engine, with equal reduction in capacity. Increasing temperature would tend to favor the turbine.

It will be noticed that at the highest pressure the engine is at its highest efficiency and the turbine at its lowest efficiency, while for low pressures the engine is at its lowest efficiency and the turbine is at its highest efficiency, but would begin to drop off at a point just beyond the limit of this chart. These curves plainly show the engine as a high-pressure machine and the turbine as a low-pressure condensing unit.

In back-pressure industrial operation the reciprocating steam unit always has been recognized as superior in efficiency to the turbine. With the introduction of higher initial pressures the superiority in efficiency will be increased, for the higher the initial pressure or the higher the back pressure, the greater the gain will be. Owing to this greater efficiency there will be many industrial plants now operating condensing or part condensing, that will be able to generate all their power as a byproduct from the steam used in the manufacturing system, and thus eliminate the condensing plant.

Assume a manufacturing system requiring steam for process or heating at 5 lb. gage pressure, in the author's opinion the reciprocating engine unit will have a lower steam rate and produce more power per given quantity of steam delivered to the system, from an initial pressure of 400 lb. than the turbine at double the pressure, or 800 lb. When working against the same back pressure of 5 lb. gage, the engine will show better results in power output with 800-lb. steam than the turbine unit will at any point up to the critical pressure of 3,200 pounds.

Losses in an engine are: Condensation or cooling of the incoming steam on the internal surface of the steam cylinders, leakage losses past the valves and pistons, mechanical or friction losses in engine bearings and working surfaces, and leakage of packing glands.

In the turbine the losses are: Leakage past diaphragms, leakage past the blades owing to clearance, windage of revolving buckets or blades, friction of steam on revolving disks, leakage of shaft glands, and mechanical or friction losses in the bearings and working parts.

In addition to the foregoing there are in each case radiation losses that are so small in proportion as to be negligible. In the engine all the losses mentioned are reduced as the pressure is increased, with the exception of the one last named, which may be slightly increased unless the packing glands are kept in perfect condition. In the turbine all the losses mentioned are increased as the pressure is increased, with the exception of the mechanical friction, which remains almost constant.

Much of the time and money that have been devoted to the lower pressures ranges or the vacuum field probably would have shown greater gain if they had been applied to the high-pressure end of the steam cycle on which we are now working. It may be said that we are just beginning a new era in steam power. The next five or ten years will produce developments in the nature of new fields for steam units, high efficiencies, low construction costs and low power costs that will be more remarkable than all the accomplishments in this line of the last fifty years.

Tempering Steel with Steam

Installation of 1,200-lb.-pressure boilers brings back for practical consideration an old suggestion that once had little more than academic interest. Reference is made to use of high-pressure saturated steam or liquid for the accurate "tempering" of steel tools. The "light straw" temper of about 440 deg. F. corresponds to a steam temperature of 365 lb. gage. The "dark straw" to "yellow brown" range (456 to 483 deg.) perhaps includes the greatest number of edge tools. The corresponding range of gage pressures is 433 to 566 lb. The highest temperature of 530 to 570 deg. F., corresponding to the dark purple or full blue of circular saws, springs, etc., could be obtained with saturated steam pressures of 868 lb. and 1,209 lb. respectively.

INTERSTAGE REHEATING

Definitely Established

By F. S. Collings

Mechanical Engineer, Sargent & Lundy, Inc.

IN DETERMINING the steam conditions for which a modern power station is to be designed, the question for or against interstage reheating is among the most prominent of the problems involved, and at the outset it is often a controlling factor in the choice between 400 lb. pressure and pressures above 400 lb. At this time it is accepted generally that the next commercial step in the pressure range above 400 lb. is 600 lb., and that at this pressure some method of interstage reheating is usually desirable.

A glance at the Mollier diagram will indicate that as progress is made along lines of decreasing entropy through rising pressures to a limit of, say, 2,000 lb., an increasing proportion of the work done by the turbine is in the saturated field. For example, in a line of adiabatic expansion from 1,200 lb. abs. and 750 deg. F. total temperature to 1 in. Hg. abs. back pressure, no less than 73 per cent of the work is done below saturation, as compared to 56 per cent in the case of a pound of steam expanding adiabatically from 400 lb. abs. and 750 deg. F. total temperature to 1 in. (mercury) abs. back pressure.

Fig. 1 illustrates the expansion line of a typical steam turbine from a throttle condition of 650 lb. abs. and 725 deg. F. to 1 in. (mercury) abs. back pressure, without reheat, with live-steam reheat to 450 deg. F. and with flue-gas reheat to 750 deg. F. It will be noticed that of the 399 B.t.u. of available work per pound of steam in the first case, 54.3 per cent is in the saturated field; and of the 436 B.t.u. available in the second case, 35.8 per cent is in the saturated field; while of the 505 B.t.u. per pound turned into work in the third case, only 20.4 per cent is below saturation. The quality of the steam at the turbine exhaust is, in the same order, 87.5, 90.7 and 93.7 per cent.

As the steam expands below saturation, the moisture that is precipitated reduces the efficiency of the turbine low-pressure blading, and in addition is a cause of erosion in the lower stages. The loss of blade efficiency owing to moisture in the steam is not less than one per cent and may reach 1.08 per cent for each one per cent of moisture present.

For pressures of 600 lb. or greater the author feels that the merits of interstage reheating from the viewpoint of higher thermal economy and lower turbine maintenance are self-evident, although in certain plants operating conditions may so affect the commercial economics that reheat may not be justified. The possibilities of the general arrangements of interstage reheating with live steam, flue gases or a combination of the two, are reviewed, with added comment on the possible use of mercury vapor or di-phenyl-oxide.

Thus in one specific case, in which the steam pressure at the turbine throttle is 1,200 lb. gage and the total temperature 725 deg. F., the decrease in the heat consumption per kilowatt of turbine output is 3.95 per cent with live steam reheat to a total temperature of 550 deg. F., as compared to no reheat, while the corresponding decrease with flue-gas reheating to a total temperature of 750 deg. F. is 7 per cent. It should be added that the gain in economy from regenerative feed-water heating is slightly less with flue-gas reheat than with live-steam reheat, owing to the increased loss of energy in the steam bled at given pressures below the reheat point, and the figures quoted take into account the difference in the available heat of the steam at the various bleed points under the two sets of conditions; the final feed-water temperature being 400 deg. F. and the absolute back pressure at the low-pressure turbine exhaust being 1 in. (mercury) in both cases.

Fig. 2 shows the improvement in economy from reheating with live steam and reheating with flue gases for 600-lb. and 1,200-lb. turbine units. It also illustrates the relation between this economy and the pressure at which the steam is reheated. The steam is expanded to one inch absolute back pressure in all cases. The performance is corrected for the effects of four-point regenerative bleeding in each case, the final feed-water temperature being 400 deg. F. in the 1,200-lb. cycle and 350 deg. F. in the 600-lb. system.

In other words, the writer feels that so far as thermal economy and turbine maintenance are concerned, the merits of interstage reheating are self-apparent when steam pressures of 600 lb. or more are to be used, but it must be recognized that from an operating point of view, as well as from that of commercial economics, there may exist conditions under which reheat cannot be justified even in a high-pressure plant.

For example, in the case of a turbine unit that is to be shut down every night, the installation of a flue-gas reheater would be likely to prove a liability rather than an asset, since the reheater necessarily must be shut down with the unit. Not only would the return on the invest-

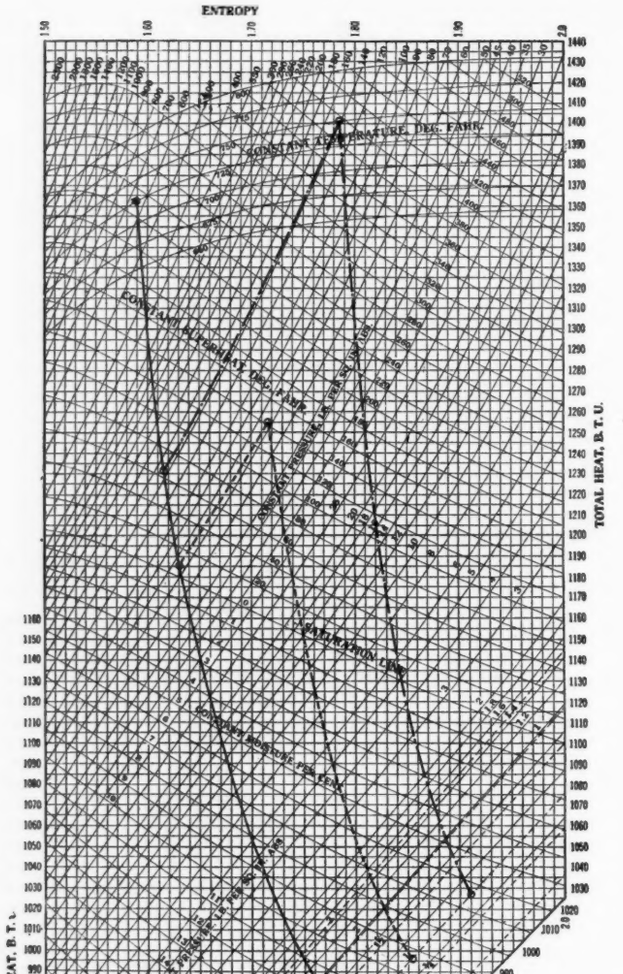

ENTROPY
TOTAL HEAT, B. T. U.
NO REHEAT
FLUE GAS REHEAT
LIVE STEAM REHEAT
Total Heat-Entropy Diagram
Data below 320 Deg. F. from Marks & Davis Steam Tables, Reprint of 1913. Data above 320 Deg. F. from Davis-Kleinschmidt Joule-Thomson Tests, Mechanical Engineering, February, 1924. Based on charts published in Mechanical Engineering, February 1926 which were computed by General Electric Company from preliminary data obtained through courtesy of the A. S. M. E. Committee on Properties of Steam and the Extension of the Steam Tables, and supplemented by data from 1200 to 3300 lb., abs., compiled by The Superheater Company from A. Stodola: Dampf und Gasturbinen, Fifth Edition, 1924.

Fig. 1—Turbine expansion curves on total heat-entropy diagram, showing percentage of work done in the saturated field without reheat, with live-steam reheat and with gas reheat

ment be small, owing to the number of idle hours in each 24, but the heat loss, from banking or heating and draining the reheater and reheat piping each morning, would be prohibitive.

Interstage reheating may be applied in any of four general arrangements:

1. Single-stage reheating with live steam only.
2. Single-stage reheating with the flue gases taken at from 1,200 to 1,500 deg. F. from the gas outlet of one or more of the high-pressure boilers.
3. Single-stage reheating with live steam and flue-gas reheaters in series.
4. Reheating with the flue gases at a relatively high-pressure stage and again with live steam at a low-pressure stage.

At first glance the simplest of these four systems is that using the single-stage live-steam reheater, since it is self-regulating as to the reheat temperature and as it takes its supply of live steam from the high-pressure steam heater and not from any one particular boiler. Against these two advantages, however, must be weighed the following limitations:

The temperature to which it can raise the steam to be reheated is limited to the temperature due to the pressure of the live steam at saturation minus the temperature difference imposed by the drop through the heating surface. For this reason it is necessary to reheat at a low-pressure stage of the turbine, about 70 lb. abs. in the case of a 600-lb. machine, in order to obtain a reasonable return from the reheater, and this low pressure involves high specific volume with correspondingly large reheaters, valves and piping connections to avoid a prohibitive pressure drop through the reheat system. Again, the disposal of the condensation from the live-steam coil of the reheater presents a difficult problem in pumping, combining, as it does, high pressure, high temperature and a low rate of flow. If, as an alternative, the condensation is cascaded through the bleeder heaters, it must be at the cost of reducing the amount of steam bled from the turbine.

As an alternative, the live-steam reheater may be located above the high-pressure boiler, so that the coil will drain by gravity to the boiler drum, but this of course necessitates a long double run of large piping between the turbine and reheater, with the attendant evils of increased investment, increased pressure drop and greater radiation loss. It should be mentioned also that the use of the live-steam reheater calls for additional high-pressure boiler surface, coal-burning equipment, etc., to supply it with the necessary steam, although the quantity of steam involved is not of such significance as to constitute an argument against this method of reheating.

Advantages of the gas reheater are that the temperature to which it can raise the steam from the turbine is limited only by the characteristics of the material available for turbine construction, further that this temperature can be regulated at will and that it does not call for increased steaming capacity in the boiler room. It does necessitate long steam piping connections between the turbine and reheater, but as the steam may be reheated to at least 750 deg. F., this may be done economically at a pressure of about 200 lb. abs. in the case of a 600-lb. plant and at about 400 lb. abs. in a 1,200-lb. plant. Hence, the specific volume of the steam is reasonably low and the reheat piping considerably smaller than must be used for live-steam reheating. Also, a pressure drop of, say, 15 lb. over-all is less serious, considered as loss in turbine pressure head, at the 200-lb. stage than at the 70-lb. stage.

Some four years of operation has demonstrated that the reheat boiler of which the gas heater is an integral part, is an easily controlled piece of apparatus, and that the variations in temperature that had been anticipated as possible, do not occur in practice. This fact is partly due to what may be called the "thermal inertia" of the reheat system owing to its mass, and partly to the fact that if the flow of steam to the reheater decreases rapidly, owing to a reduction of the load on the turbine, the total heat per pound of the steam passing from the turbine to the reheater decreases also, so that each pound of

steam may absorb more heat in the reheater without increasing its final temperature unduly. For example, in one typical 600-lb. installation, the heat transferred in the reheater per pound of steam at half load was 139 per cent of that transferred at full load while the total heat (Pounds of steam × B.t.u. rise) transferred at half load was 66 per cent of that transferred at full load.

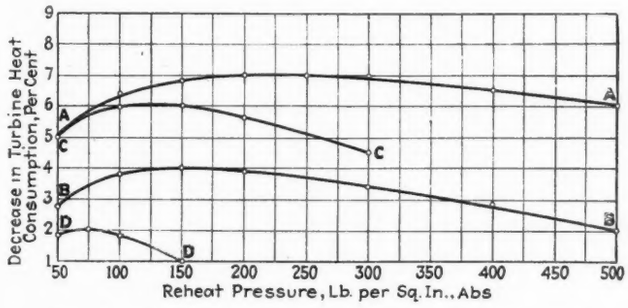


Fig. 2—Gain in economy from reheating at various pressures

A-A, 1,215 lb. abs., 725 deg. F. at throttle—Reheat to 750 deg. F. with flue gases.
B-B, 1,215 lb. abs., 725 deg. F. at throttle—Reheat to 550 deg. F. with live steam.
C-C, 615 lb. abs., 725 deg. F. at throttle—Reheat to 750 deg. F. with flue gases.
D-D, 615 lb. abs., 725 deg. F. at throttle—Reheat to 450 deg. F. with live steam.

Fig. 3 illustrates the steam and feed-water cycle of such a representative installation, having single-stage gas reheat. The gas reheater, however, is a bulky piece of equipment, and so long as it is arranged as at present, as an integral part of a unit comprising a small section of high-pressure boiler surface, a high-pressure superheater, the reheater, an economizer or air preheater, and a furnace of adequate volume, its capacity must be kept within a definite limit unless the reheat boiler dimensions are to exceed those of the standard boilers, which would be undesirable from both the physical and economic viewpoints. As an alternative the steam from the turbine may be divided between two or more reheat boiler units, but with the increasing size and decreasing steam consumption per kilowatt of turbine units this arrangement may lead to all the boilers pertaining to each turbine unit being reheat boilers. When this point is reached, the engineer will have a choice of three courses.

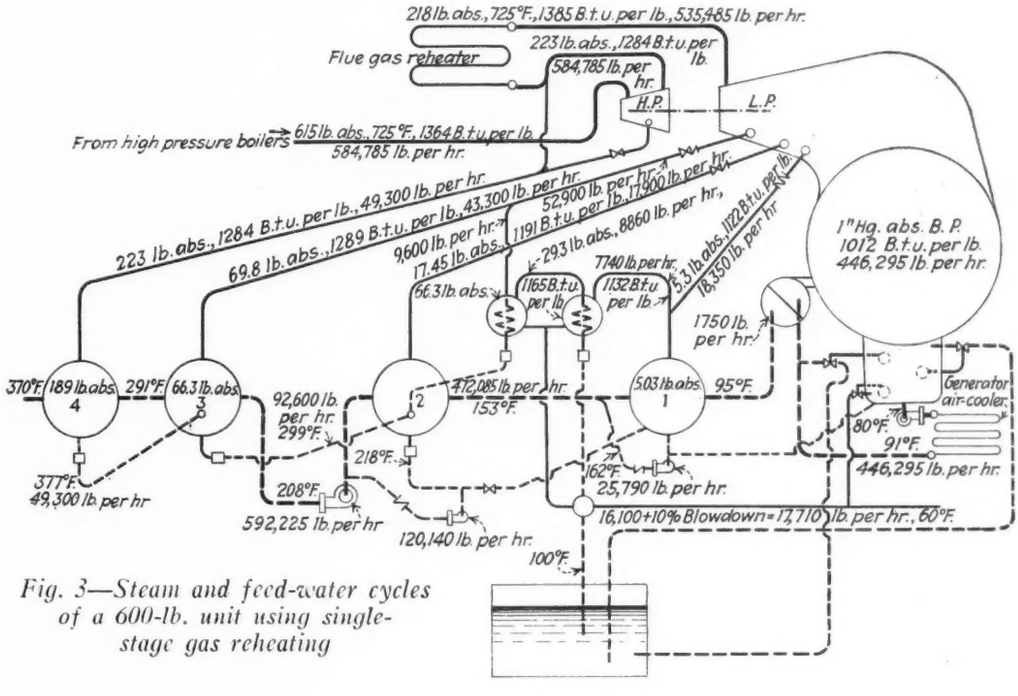


Fig. 3—Steam and feed-water cycles of a 600-lb. unit using single-stage gas reheating

He may proportion his boiler-room equipment so that when generating sufficient high-pressure steam for the full load on the turbine, the reheater will give the maximum allowable temperature to the reheated steam. He must then suffer some loss in economy at the lighter loads owing to reduction in reheat temperatures at lower boiler ratings.

He may install live-steam reheaters in series with the gas reheaters, the former to be idle at high turbine loads, but to help the latter to maintain full reheat temperature at fractional loads by relieving them of some part of their work.

The third choice would be to adopt two-stage reheating. Thus, with a 1,200-lb. installation, for example, the steam from the high-pressure cylinder would be reheated at about 400 lb. pressure in the flue-gas reheater, one of which is installed in connection with each boiler, and reheated again at 60 or 70 lb. pressure in the live-steam reheater. The latter, of course, would maintain its exit temperature to within a few degrees of 550 deg. F. at all loads and, in maintaining the desired condition of the steam in the low-pressure cylinder, would counteract to a great extent the adverse effects of the reduced superheat in the intermediate cylinder.

Of the three schemes outlined previously, the second and third involve increased investment for the live-steam reheaters, valves and piping, and especially in the case of the second plan, as probably it would be advantageous to place the live-steam reheaters close to but above the gas reheaters with which they are in series, so that the condensate will drain by gravity to the boiler drums.

Interstage reheating with units of large capacity is simplified to a great extent by the adoption of the three-cylinder cross-compound turbine. With this type of machine the exhaust steam from the high-pressure cylinder is divided between the two low-pressure cylinders, and so the work of reheating may be divided conveniently between two reheat boilers without the necessity of operating these boilers in parallel with respect to their reheat sections. If, however, the engineer's choice or the force of circumstances directs that the two-cylinder tandem-compound single-shaft type of turbine unit be used for capacities of 100,000 kw. or more, the problem of reheating, as before stated, does involve difficulties on account of the physical dimensions

of the reheaters and of their control when divided into a number of units to be operated in parallel.

In this connection it is necessary to mention one important factor, that of the division of the total kilowatt capacity of the unit among its generators. It is of considerable advantage, for physical reasons, to reheat at the high-pressure cylinder exhaust, rather than extract at an intermediate stage for reheating, return to the same cylinder, and then use a crossover connection to the throttle of the low-pressure cylinder. This fact being accepted, the relative capacities of the high- and low-pressure generators will establish the reheat point, and the stage pressure at this point will influence or even decide the choice between flue-gas reheat and live-steam reheat.

All the foregoing facts, figures and conclusions are based on present-day practice and on the premise that it is not desirable at this time to exceed 750 deg. F. as a working steam temperature.

It is possible that at some future date interstage reheating will be accomplished by the use of closed tubular heaters somewhat akin to the live-steam reheater and located adjacent to the turbine, but using mercury vapor, di-phenyl-oxide, or some similar agent having characteristics permitting high vapor temperatures at relatively low pressures.

Of the two agents mentioned, it is probable that the selection would be in favor of mercury vapor, since the properties of di-phenyl-oxide appear to be unstable at elevated temperatures, but the use of mercury vapor also presents a number of grave problems. As a hypothetical case it may be assumed that a machine capable of operating at a reheat temperature of 850 deg. F. is within the possibilities. The total temperature of mercury vapor at 80 lb. abs. and saturation is 874.8 deg. F., which would allow of a reasonable terminal difference, and inasmuch as the turbine reheat steam pressure would be at least double that of the mercury, a leak between the heating surfaces would not discharge mercury vapor into the turbine. However, the total heat per pound of mercury vapor at 80 lb. abs. saturated is 149.2 B.t.u. and the heat of the liquid 31.4 B.t.u. The heat of vaporization, then, is only 117.8 B.t.u. per pound.

A 1,200-lb. 725 deg. F. turbine reheating at 400 lb. abs. to 850 deg. F. would require an addition of heat in the reheater of 185 B.t.u. per pound. Such a unit of 100,000 kw. capacity and using the regenerative feed-heating cycle to a final feed temperature of 375 deg. F. would pass through the reheater 766,000 lb. of steam per hour and would require, therefore, 1,203,000 lb. of mercury vapor per hour at 80 lb. abs. in order to produce a temperature of 850 deg. F. at the reheater outlet. One of the first problems, therefore, would be to develop a system of forced circulation in the mercury boiler so that neither its physical dimensions nor its mercury content would be prohibitive, and in this connection the great difference in mercury pressure at different depths must be remembered.

A second problem is presented by the fact that the mercury boiler must be separately fired, for the reason that as mercury does not wet a steel surface, a high temperature difference is necessary in order to produce rapid boiling. Again, it is obvious that the steam reheater, or mercury vapor condenser, must be located so that it may drain by gravity to the mercury boiler. Of the difficulties mentioned the first is the greatest, since the size of the mercury boiler and the weight of mercury must be such that the investment charges will not offset the increased economy from high-temperature reheating for large turbine units.

However, speculations of this nature, while interesting as a study, are not of immediate concern to the engineer whose business it is to obtain the best results under existing conditions, and with the material now available albeit with one eye on the future.

Interstage reheating has become a factor of great importance in the improvement of operating efficiency and the reduction of turbine maintenance costs and, except under unusual circumstances, cannot be ignored in the design of high-pressure steam-driven generating stations. Its application may and does vary with local conditions, but as a principle it is most definitely established.

Dependable Heat Insulation For High Temperatures

The problem of heat insulation for high steam temperatures is somewhat more complex than that for lower temperatures, but it has been commercially solved for temperatures considerably higher than the 750 deg. F., which is the present usual limit of steam temperatures.

For temperatures below 600 deg. entirely satisfactory service is given by 85 per cent magnesia and suitable types of laminated asbestos covering. Temperatures much higher than 600 deg. are injurious to eighty-five-per cent magnesia and also to the laminated coverings. In the first case the carbonate of magnesia starts to calcine to the oxide at about 550 deg. The calcined magnesia is a first-class heat insulator, but has little mechanical strength. However, the temperature falls steadily through the insulation from the inside to the outside so that the weakening is confined to that inner region where the temperature exceeds 550 deg. Thus if the covering is left in place and not subjected to severe shock or vibration it can undoubtedly give long and efficient service with steam temperatures well above the point where calcining starts.

While asbestos is not, as some imagine, unaffected by heat, the weakening of the fibre is not serious at temperatures up to 800 deg. All asbestos insulations, however, require binding media. In general the binders that best withstand heat are not good heat insulators. Thus the most efficient asbestos insulations are subject to approximately the same limitations as 85 per cent magnesia.

To meet this situation special high-temperature insulations, in sectional, block and sheet form, have been placed on the market. They are being used widely at temperatures up to 750 deg. and in oil refineries at temperatures as high as 1,000 deg. These high-temperature insulations are somewhat less efficient than 85 per cent magnesia or laminated asbestos covering. In addition they usually cost more. This explains the custom of using the high-temperature insulation as an inner layer only, surrounding it with a layer of the more efficient low-temperature insulation.

Theoretically the inner layer should be merely thick enough to protect the outer from temperatures higher than 600 deg. In practice, however, the minimum thickness of the inner layer is limited by mechanical and manufacturing considerations. It is generally at least $1\frac{1}{2}$-in. thick, except for small pipe.

Determination of the correct total thickness is based on the established economic principles as applied to low temperature insulation, but the computations required are somewhat more difficult.

PIPES *and* JOINTS

This lap joint on an 8-in. pipe withstood a pull of 720,000 pounds

For High Pressures and Temperatures

By A. W. Moulder
Manager, Heating and Power Piping Division, Grinnell Company

INCREASED steam pressures and temperatures have presented new problems in piping for the manufacturer of basic materials, for the fabricator and for the engineer designing the piping system.

The requirements in pipe have been adequately met by the establishment of the A.S.T.M. standards requiring seamless tubing exclusively in varying thicknesses and weights for pressures of 600 lb. and above and requiring adequate thicknesses of lap-welded pipe and seamless tubing for lower pressures.

Likewise, in fittings the A.S.T.M. tentative standards for steel fittings have been fully developed for steam pressures up to and including 1,350 lb., and manufacturers are regularly furnishing products to these standards.

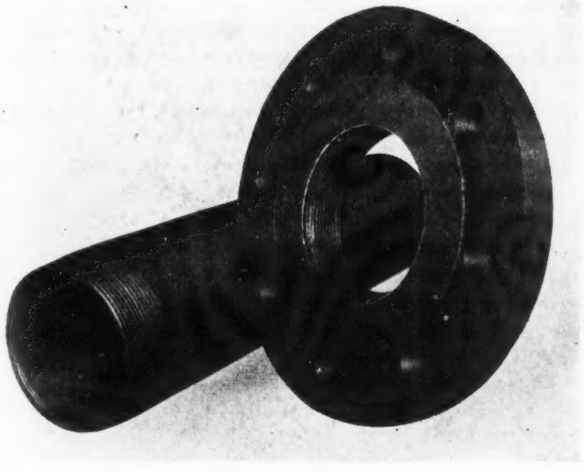


Fig. 1—This threaded joint pulled out at a load equal to 59 per cent of the strength of the pipe material

In pipe these A.S.T.M. standards are being almost universally followed but one unfortunate condition still exists in that seamless tubing in the larger sizes is not as yet being produced in this country on a commercial basis competitive with imported tubing. Therefore, for this service most of the tubing in the large O.D. sizes is being imported, with consequent delays and other unsatisfactory conditions. Fortunately, this situation will probably not endure for any great time, as a number of American manufacturers are increasing their facilities along these lines.

With pipe joints little has been done, or for that matter can be done, to develop a standard, because of the varying conditions encountered or developed in erection conditions.

Theoretically, welding appeals to almost anyone as the near-perfect answer to this pipe-joint problem. The mental picture of a piping system which, when finished, has every joint welded into one continuous piece, with branches and straight runs of piping all continuously welded, is certainly most alluring. Its appeal arises from several natural advantages:

(*a*) Permanent tightness of joints.

(*b*) Economy of first cost (in many cases).

(*c*) Improved service in installation, because of simplicity of material requirements.

(*d*) Improved efficiency and decreased cost of insulation, because of relative smoothness of finished piping system.

In boiler plants and power plants operating with pressures up to 250 lb. (many with some superheat) it will be found that there has been an ever-increasing amount of welded piping used. Welded headers have become gradually the rule rather than the exception. A number of plants have been installed with practically all of the piping welded. It is safe to say that all these installa-

tions, *where the welds were made by welders properly trained in pipe welding and with the work done under proper supervision,* have been practically 100 per cent satisfactory.

There are no definite indications that the welding process cannot keep pace with the requirements in materials for increased pressures and temperatures. In fact, there is considerable experience indicating the contrary. One thing helping toward this end has been the development of high-strength filler-rod materials. Those in common use now for important high-pressure service average as high as 65,000 lb. tensile strength, and rod is available in

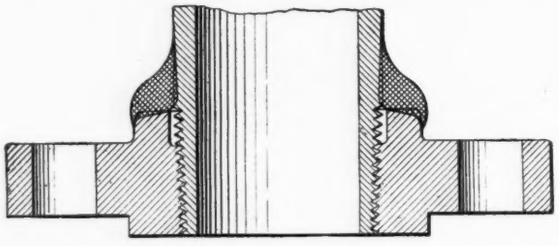


Fig. 2—Steel flange screwed on and welded

limited quantities with a tensile strength of 85,000 lb.

Welding for increased pressures and temperatures has been used largely in conjunction with boiler headers or distributing headers for power plants. These are made by any one of three processes: 1. Forge and hammer welding; 2. Oxyacetylene welding; 3. Electric-arc welding.

Of the three, No. 1 has been looked upon with most favor for the higher pressures. Probably this is because it is the oldest of the three processes. Inasmuch as all three are so greatly dependent upon the human element for successful results, the oldest and best known of the processes would be looked upon with the greatest assurance. It has been possible, however, to place greater dependence upon the supervision or management over the No. 1 process because of the relatively greater investment in equipment required and, therefore, the relatively fewer and more responsible fabricators offering such welding. It has been difficult to specify requirements for autogenous welding so that proper supervision and management over the human element would be insured, and such insurance is essential to dependable performance.

Experience, however, with welded headers made by any one of the three processes, when the work has been done by trained men under proper management has, so far as the writer can learn, been 100 per cent satisfactory. These headers have been quite common up to 400 and 600 lb. pressure and in the forge-hammer-weld type for higher pressures.

Of course, practically all of these headers have been made under excellent conditions in that they have been fabricated in the shops of reliable pipe-fabricating concerns, where the best facilities were available for preparation of the materials, skilled workmanship under good management, annealing and finally, proper test.

The test for welded piping as prescribed by the specifications of the Power Piping Society and as regularly practiced by most fabricators requires:

"Wrought pipe which has been welded by the fusion process shall be tested to the following pressures per square inch:

600 lb. W.S.P. rating	1,500-lb. test
400 lb. W.S.P. rating	1,000-lb. test
250 lb. W.S.P. rating	750-lb. test
125 lb. W.S.P. rating	300-lb. test

The welded pipe shall be struck near the weld, while under pressure, with a two-pound hammer or equivalent."

It is by this test that the purchaser can best safeguard himself in buying welded headers and similar products.

The welded header of fifteen years ago for low and medium steam pressures was the forerunner of a much broader use of this process in power-plant and industrial steam piping. Whether the welded header used with the pressures and temperatures of today will become the ancestor of a similar broadening of the use of this process remains to be seen.

No matter how welding may develop, we have with us and will always have with us to a great extent the necessity for dependable flanged pipe joints to take care of certain features in layouts where welding is not practical or economical. Considerable progress has already been made in producing flanged joints that are dependable with increased pressures and temperatures.

Many years before present-day requirements had been reached, it was realized that the screwed cast-iron flange did not represent the ideal because cast-iron "grows," with expansion, out of proportion to the steel pipe. This finally leaves a space between the two, resulting in a leaky joint. Even with the later advent of steel flanges the screwed flange joint did not represent the ideal. Fig. 1 shows the result of a typical pulling test with a screwed flange. The threads in this case were pulled out at a load of 99,080 lb., or at approximately 59 per cent of the ultimate strength of the pipe material itself.

To overcome thread leaks the best development up to the last few years was the commercial type of lap joint of today, illustrated in Fig. 3. Fig. 4 shows the result of a typical pulling test of this type of joint. The lap joint is pulled through clean at a load of 96,670 lb., or approximately 58 per cent of the ultimate strength of the pipe material itself.

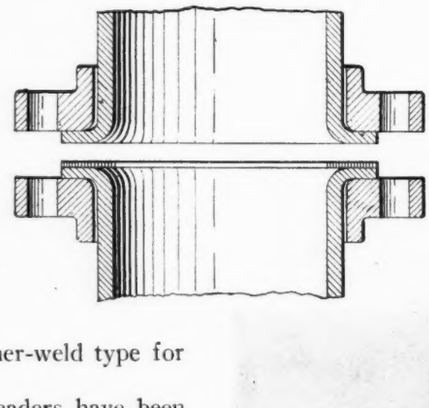
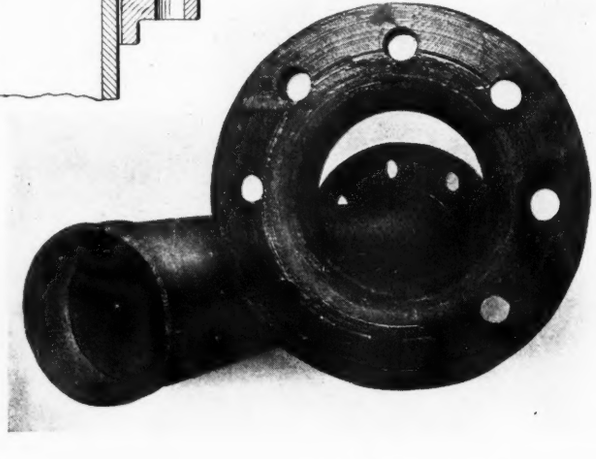


Fig. 3—Commercial type of lap joint without reinforcement

Fig. 4—Non-reinforced lap joint pulled through the flange at a load equal to 58 per cent of the strength of the pipe material



Obviously, what was wanted was a type of joint as strong as the pipe material itself. The lap joint has apparent advantages in assembly over screwed joints as well as obviating the leak troubles of the latter; therefore, it was logical that any development in pipe joints for higher pressures and temperatures should be in lap joints. This meant the design of heavier machinery for upsetting the pipe ends while rolling lap joints, and at the same time faster machinery to insure that the heavier laps could be rolled in the same time, so that the work would be done at the correct temperature.

The result has been that certain pipe fabricators have developed processes for making lap joints which, when completed in their rough state, will permit:

(*a*) Machining on back of laps, to insure even bearing of flange over entire area.

(*b*) Machining of various types of faces.

(*c*) Finally that, after all machining is done, the minimum thickness of laps at any point will be equal to or greater than the pipe-wall thickness.

The photograph at the head of the article is a view of a lap joint on the end of an 8-in. seamless pipe for 600-lb. working pressure. This is equipped with a 900-lb. steel flange. As shown, it has withstood a pulling load of 720,000 lb.

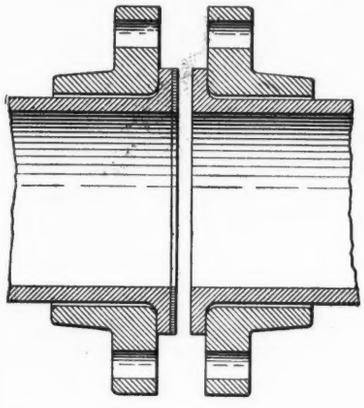


Fig. 5—Heavy-duty, plain-face lap joint

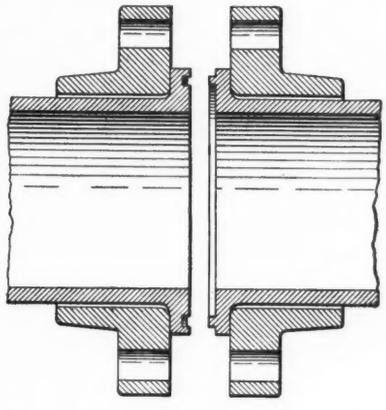


Fig. 6—Heavy-duty, tongue-and-groove lap joint

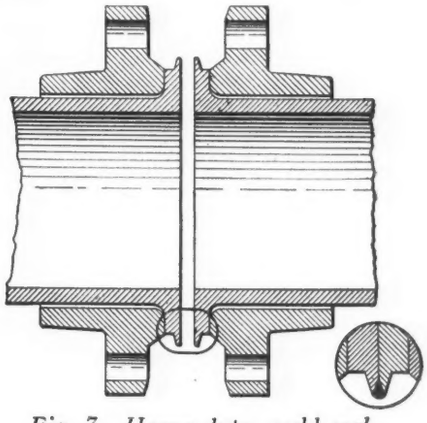


Fig. 7—Heavy-duty, weld-seal joint

As shown by Figs. 5, 6 and 7, various types of facings are obtainable from these heavy upset types of lap joints, and therefore the designing engineer is given a wide choice. Each type has certain advantages and disadvantages, as indicated by the following:

The heavy-duty, plain-face lap joint, shown in Fig. 5, is the least expensive of the three. It also attaches to stock types of valves, fittings and flanges of boilers, etc., without special facings.

The square corner at the inside covers the full face of the gasket, preventing cutting off with consequent troubles in valve seats, etc.

An important advantage of the heavy-duty, tongue-and-groove lap joint shown in Fig. 6, is that the gasket (usually soft annealed copper) is fully protected against blowing out. Another advantage lies in the fact that the unit pressure exerted on the gasket by the bolts is much greater than on gaskets of larger area. This type of joint is somewhat more expensive than that illustrated by Fig. 5. It has the disadvantage of requiring special facing of flanged fittings, valves, boiler nozzles, etc., and exercise of care to be sure that the proper flange (either tongue or groove type) is always ordered so as to match.

The heavy-duty "Sarlum" or weld-seal joint shown in Fig. 7 combines the assured safety of the well-standardized flanged joint with the permanent tightness of the all-welded installation. Because of these features it is used in many of our largest public-utility power plants operating under very high pressures and temperatures.

The total installed cost of this type of joint is usually greater than that of either of the other two types.

It has the disadvantage of necessitating also the special facings on adjacent valves, fittings, nozzles, etc., although this disadvantage is not as pronounced as with the tongue-and-groove-type joint.

One other type of flanged joint will no doubt occur as a possibility for use in high-pressure and high-temperature service. This is the steel flange screwed on and welded as shown by Fig. 2. This provides a substantial job, but is usually applied in the smaller sizes only. It has a disadvantage in that if sufficient welding is added to increase the strength of the threaded joint materially, a warping effect is usually produced. This, of course, is more pronounced as the sizes increase. It also has an apparent disadvantage in that the bolt holes must be lined up exactly to fit the adjacent flange prior to welding. The lap-joint types of flanges, owing to their swivel construction, overcome this installation disadvantage.

There are a number of other items, the importance of which has been magnified in direct proportion to recent increases in pressures and temperature. Chief among these are:

(*a*) Care for expansion and contraction.

(*b*) Supporting or hanging.

Item (*a*) is one which in many cases goes right into the building design itself, because under present conditions we are called upon to take care of much greater movement of piping lines in a much more limited area. The solution is usually different for every installation, but the designing engineer does well to be extra safe in the design of anchors and free with the use of bends or turns in the piping. Available data on this subject are all too meager.

Item (*b*) would often be left to the installation foreman under old conditions, but now appears as one of the major problems in the design of the piping system. Next to the importance of safety comes the importance of adjustment. Many of our proposed new tentative A.S.M.E. piping codes provide that "hangers for all piping 3½ in. and larger shall be capable of screw adjustment after erection." This is a good provision, because it is important that after erection each hanger or support should be adjusted to carry its share of the load.

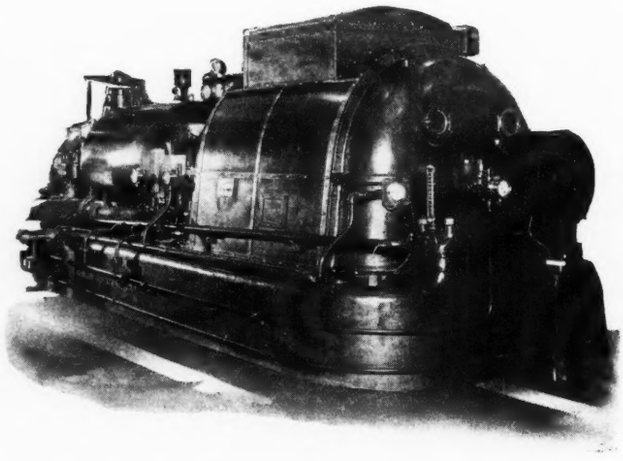
Operating Experiences

A symposium in which I. E. Moultrop, M. K. Drewry and J. C. Segeler speak from first-hand knowledge of high-pressure operation

"THE proof of the pudding is in the eating." No amount of theoretical promise will satisfy the average engineer that he may safely consider a high-pressure installation. Hence this symposium of operating experience, in which the engineers of pioneer high-pressure installations report their troubles—surprisingly few—and tell why they see a great future for the high-pressure plant.

Irving E. Moultrop, Chief Engineer of the Edison Electric Illuminating Company, Boston, bases his report on experience at Edgar Station, America's first 1,200-lb. installation. M. K. Drewry, Assistant Chief Engineer, Power Plants, Milwaukee Electric Railway & Light Company, predicts a bright future for high pressures on the basis of actual experience with 1,300 lb. at Lakeside Station.

Industrial plants are represented by J. C. Segeler, who describes the 1,000-lb. installation of the Masonite Corporation and reports on three years' operation.



One of the 1,200 lb. pressure turbines at the Edgar Station

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Commercial Economy Proved at Edgar

By IRVING E. MOULTROP

Chief Engineer, Edison Electric Illuminating Company of Boston

THREE and one-half years ago the first commercial 1,200-lb. installation in the world was put into operation in the Edgar Station of the Edison Electric Illuminating Company of Boston. That installation has definitely proved the commercial economy of 1,200-lb. equipment for steam-electric generating stations.

Many operating difficulties were looked for with the first installation at this radical increase in pressure, but fortunately those difficulties that were expected failed to materialize. The few minor troubles that were experienced have been discussed in the technical press. None was of such a nature as to cause any doubt about the success of the project. The first 1,200-lb. boiler and turbines are today operating satisfactorily.

The first installation, as was to be expected, was not the most efficient either thermally or in the use of materials. However, a study of the problem after an extended period of operation has shown that the expected economies have been realized and the costs of construction were such as to justify further installations at this pressure.

We can, therefore, now state definitely that equipment for pressures around 1,200 lb. has passed from the experimental stage into a stage of intensive development. Later designs show a marked improvement both thermally and in the use of materials and have resulted in a reduction in the cost of construction per unit of capacity. The capacity of the steam-generating units has been doubled, while the pressure parts have been more economically distributed between boiler and economizer so as to result in a better unit from both the design and the operating points of view. Turbine-generator units have been built with four times the capacity of the original unit and, by the use of multiple valves, are able to operate at good economy over wide ranges in load.

There are in service today three stations operating equipment at approximately 1,200 lb. pressure and five others are under construction.

In reference to the use of higher steam temperatures we have known for many years that higher temperatures were desirable, but the materials of construction have held back the adoption of radically higher temperatures. Experience with cast-steel pressure parts has not been altogether happy, and we have been of the opinion that higher temperatures would require the use of alloy steels for those parts subjected to the high temperatures. We have, therefore, proceeded slowly in raising the operating temperatures, although temperatures have been a little more than doubled in the last thirty years. A vast amount of research work has been done to produce commercial materials that would permit the use of higher temperatures, but all that is to be desired has not been achieved. Recent betterments in design, however, have permitted the use of somewhat higher temperatures with the materials available, and we can, therefore, expect some increase in operating temperatures in the near future.

What is the reason for this trend toward higher pres-

With High Pressures

sures and temperatures? The reason is the desire to reduce the total cost of generating electrical energy. It is very gratifying that higher thermal efficiencies have been obtained, but the actuating motive has been the desire to reduce the total cost of electrical energy and not a desire for thermal efficiency records. To put it another way, we are past the period in our development when an engineer can select a particular type of equipment to please his own likes or dislikes. An engineer's business is to design and operate so as to furnish the quality of service demanded at the lowest rate commensurate with the money invested.

There have been a number of periods in the history of the electrical industry when it has appeared that many power-plant engineers were installing certain equipment just because someone else had successfully done so, without reference to the economic justification of the equipment involved. We are doing better engineering today than in the past, and it is improbable that the present trend toward higher pressures and temperatures is due to a similar situation. A number of engineers have made careful studies to determine the most economical operating steam conditions, and these studies have practically all agreed that for the average conditions encountered in the production of electrical energy from coal in the United States, the most economical condition today is somewhere between 1,200 and 1,500 lb. pressure and between 750 and 800 deg. F. total steam temperature. It is possible that as we raise the operating temperature the most economical pressure will rise also.

There is a very important factor, however, that should not be lost sight of, and that is the fact that in building

Fig. 2—Headers of new "Masonite" high-pressure boiler
Note thickness of header and special grooved construction to allow effective rolling in of tubes.

and operating high-pressure and high-temperature stations a high degree of engineering skill is required. The high-pressure and high-temperature parts of steam stations are expensive, and therefore the greatest possible capacity should be obtained from those parts. This is the reason for the present designs where the boilers proper are operated at heat absorptions of from 25,000 to 30,000 B.t.u. per sq.ft., and heat-reclaiming apparatus is installed in the boiler-exit gas passages to the degree that is justified by the particular station conditions. Unless the design of the station and apparatus is correct, a high-pressure station can lose money just as surely as a properly designed one can make money for the company which owns it.

To summarize, we can say that operating experience has shown that the expected gains in economy due to higher pressures and temperatures have been realized. Construction experience has shown that such installations can be made at a cost that justifies their installation. We can, therefore, say that the commercial economy of higher pressures and temperatures has been demonstrated

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Thirteen Hundred Pounds Pressure Proves Itself

By M. K. DREWRY

Assistant Chief Engineer of Power Plants
Milwaukee Electric Railway & Light Company

THE ultimate test of power-plant equipment—the ability to carry full load continuously when badly needed—has been passed successfully by 1,300-lb. equipment the past winter. An unexpected large increase in system load required continuous performance throughout the winter of the high-pressure installation at Lakeside, Milwaukee, and it carried its full share of the increase.

Proving this continuity of service are data of the recent year from April, 1928, to April, 1929, showing the boiler in service 87 per cent of the elapsed hours. On two occasions it has operated four months continuously under pressure.

For the last eleven months the 1,300-lb. turbine has been available for operation at all times. A second high-pressure turbine has been found to have 78 per cent efficiency by a weighed water test.

That the major boiler and water-wall circulation problem is not one of encouraging circulation by obtaining more and larger steam bubbles, but is rather a problem of disposing of excess amounts of steam in the risers, is now quite generally conceded. In nearly all systems the differential head inducing circulation is high, but in many the resistance head retarding circulation in the risers is also high, and circulation in the downcomers is low at all ratings. Many have observed that circulation in the downcomers actually decreases with rating. Increasingly more importance is attached to the value of high mass flow in tubes and less to high velocity flow. High pressures encourage high mass flow, thus explaining why water-wall systems are possibly more reliable at high pressures than at low ones.

With high pressures enters the practicability of higher boiler ratings. One has only to refer to the futility of operating a low-pressure heating boiler at 300 per cent rating to agree with this point. Experience in several high-pressure plants has proved better water-level control than ever previously experienced. At 1,300 lb. pres-

sure, steam weighs three pounds per cubic foot, which is much nearer the density of water than is 300-lb. steam weighing two-thirds of a pound per cubic foot. At the critical pressure, 3,226 lb., water and steam alike weigh about nineteen pounds per cubic foot.

The only exacting feature of high pressure emphasized by operating experience is the necessity of careful corrosion prevention. Whether the tendency of black oxide formation is entirely induced by oxygen in the feed water or is the result of direct union of steam and slightly overheated metal, is still unanswered. In any case oxygen in the feed water should be kept at nearly zero continuously, and mass flow in water tubes receiving heat at high transfer rates should be high. In original designs water-circulation rates proportional to heat transfer rates must be provided. Thirteen-hundred-pound boilers, in installations with high heat-transfer rates induced by small furnaces and limited heat absorption therein, will need excellent circulation.

In conclusion, one can say that 1,300-lb. pressure has qualified in the test of commercial practicability. It is positively past the trial stage. The large number of proposed and projected high-pressure installations is another proof of this point.

*

Experience of Masonite Corporation With 1,000-lb. Boiler

By J. C. SEGELER
Power Engineer, Masonite Corporation

W. H. MASON, vice-president of the Masonite Corporation, Laurel, Miss., a few years ago conceived the idea of exploding wood by means of high-pressure steam, the wood fibers to be reunited again to form insulating and pressed boards.

Today the mill is operating on a production basis of seven million square feet of board per month. The products are sold under the trade name of "Masonite." Saw- and lumber-mill refuse and those trees not considered of value to the lumber industry form the chief source of raw material for the Masonite products. For this reason the process is considered of vast economic importance.

The present steam-producing installation consists of one high- and two "low-pressure" boilers, the latter supplying steam at 175 lb. pressure for use in the treating, pressing and drying of the boards after they have been formed on the "wet-lap" machine. Condensate from the presses is returned to storage tanks to be used as feed water for the high-pressure boiler. The high-pressure boiler supplies saturated steam at 1,000 lb. to the six "guns" wherein wood chips are exploded into long fibers, these fibers to be treated, formed into sheets, pressed and dried to make the finished product.

Built by Babcock & Wilcox Company, this boiler has a total heating surface of 3,880 sq.ft. It has a single cross-drum, 4¼ in. thick, at the front. The main tubes are $\frac{5}{16}$ in. thick, and the circulating tubes, $\frac{3}{8}$ in. The peak load at times is 400 per cent of rating, the average rating being 250 per cent.

As is the case with the low-pressure boilers, wood refuse is burned under the high-pressure boiler. Natural draft is now employed, but proposed additions to the manufacturing capacity will necessitate the installation of a forced-draft blower. The peak rating at that time may be 600 per cent. No furnace-wall cooling is used at present.

No operating difficulties, outside of limited time for repairs, have been experienced. The plant operates six days a week, allowing very little time for necessary maintenance work. As there are no spare boilers to rely on, thorough inspection and prompt attention to each minor defect must be paid. The boilers have been in service for 33 months and no major repairs have had to be made.

With the high ratings carried, close control of the quality of the feed water at all times is essential. Feed water for the low-pressure boilers is taken from the deep-well supply tanks. This water is slightly acid, having a pH value of 6.1. The feed supply is pumped through a condenser located on top of the high-pressure feed-water storage tanks, where the flash vapors from the trap returns are condensed. The warm low-pressure water then passes to the feed heater where it is heated to the boiling point by the exhaust from the boiler auxiliaries. This heating removes the temporary hardness.

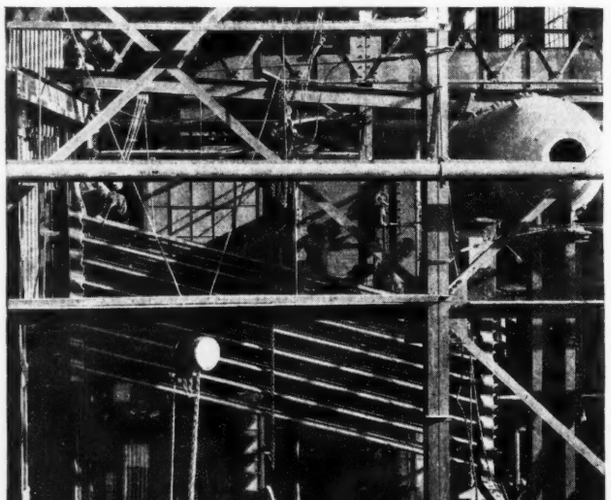


Fig. 1—New "Masonite" high-pressure boiler
The forged drum is 4⅜-in. thick and weighs 22½ tons.

A special salt solution is fed into the feed-water suction line. Periodic analyses of the low-pressure boiler-water control the amount of chemical used. A ratio of sulphate to sodium alkalinity of three to one is the standard aimed at to eliminate possibility of caustic embrittlement. All boilers are blown down once every eight hours.

At present the high-pressure feed water needs no conditioning, since it is practically pure distilled water. The feed water is kept at boiling temperature in the storage tanks, which effectively removes entrained gases. There is a small amount of leakage from the flash vapor condenser, but not enough to cause any trouble.

Before last July treated well water was used on the

high-pressure boiler also. Considerable trouble was experienced with scale and occasional tube ruptures. For this reason the feed-water system was changed so as to use condensed low-pressure steam only.

The high-pressure boiler has been in service since last July without requiring tube cleaning. A recent inspection by the insurance company was passed in fine order. Weekly inspection by our boiler foreman, M. W. Powers, is made on all the boilers. Scale must not be allowed to form in our case because of the high pressure and rating.

From an operating point of view the character of the high-pressure load is unfavorable. Steam flow fluctuates from rating to 300 per cent rating in cycles of about two minutes. The corresponding pressure drop at times is as high as 200 lb. However, no leakage of the tubes due to this condition has been experienced. Every Sunday night before the boiler is fired, a water-pressure test is applied and generally the tubes show no signs of leakage. The few times that leakage during the test has been encountered can be attributed to too rapid cooling of the boiler after shutting down on Saturday night.

The steam circulating tubes are the only ones that start leaking at these times. Unequal rate of contraction between these tubes and the other parts of the boiler is undoubtedly the cause. Gradual cooling is important because of the high temperature and metal thickness.

At the present time the high-pressure boiler pumps are of the hydraulic duplex type. Copes feed-water regulators have been ordered to replace present manual control.

The commercial efficiency of the whole boiler plant is much higher than is usual in this type of installation. The boilers consume the wood refuse not suitable for the manufacturing process. When the wood-refuse supply is insufficient, coal is fired on the front part of the grates of the two low-pressure boilers, the available wood refuse being used on the rear half of the grate surface and under the high-pressure boiler.

Draft gages, flow meters and CO_2 recorders have been installed to maintain high furnace efficiency. The average CO_2 when using wood fuel alone is well over 16 per cent, and drops only during fire cleaning periods. When wood and coal are burned together, an average of 15 per cent CO_2 is maintained. These are practical limits on our equipment.

Additions to the manufacturing capacity and the need for spare boiler capacity have required extensions to the boiler room. At the present time we are installing another Babcock & Wilcox high-pressure boiler and two 8,000-sq.ft. "low-pressure" Stirling boilers. All three are equipped with Westinghouse stokers for burning coal, although wood refuse may also be used later if available.

We consider our plant of unusual significance in that the high-pressure steam is used for process work only. There are possibilities for our own power production, also, but for the present, because we have other problems of greater importance, power generation is not being considered.

It has been our experience that a high-pressure boiler installation presents no difficulties if proper attention is paid to those details which should be considered of extreme importance in all boiler-room operation regardless of pressure.

Control of quality of feed water, proper control of water level, cleanliness of heating surface inside and outside, intelligent and regular inspection, and rigid adherence to all rules governing boiler-room operation are more of a necessity in high-pressure work, but should be considered in any case where efficient and continuous, smooth operation is expected.

While the contributors to the foregoing symposium are exponents of high pressures, it is recognized that moderate pressures are championed by a considerable number of engineers of recognized standing. Several were invited to express their views on the subject. In response to this invitation Mr. Brydon points out certain factors that justify the plant of moderate pressure.—EDITOR.

Local Conditions Influence Selection of Operating Pressures

BY H. BOYD BRYDON
Byllesby Engineering & Management Corporation

THE question of the operating pressure for new power plants appears to have attained the position of a major political issue. Those of us who, for reasons that seem economically sound, are building plants of moderate pressure, have been more or less criticized as "back numbers" and wilfully blind by those gentlemen who have built plants of high pressure; and those who have built plants of moderate pressure have undoubtedly considered the expenditures for the high-pressure plants as unwarranted.

All honor is due to the men who have been responsible for initiating the development of the high-pressure plant in the effort to achieve a reduction in the cost of electric power to the ultimate consumer. At the same time the use of high-pressure steam is not a panacea for all the ills the power house is heir to.

The selection of operating pressure for any power house is purely one of economics and must be solved by a consideration of the local conditions of capital and operating costs. On this we are all agreed. Where we differ appears to be prinicipally in estimating the extent to which the equipment of the new plant will be utilized over the period of years representing the effective life of the individual units installed in the plant, and this again is a matter for the mature judgment of the executives and their engineers.

As a general proposition it may be that the interconnection of large systems may result in a reduction in the proportion of stand-by equipment in the individual power houses, thereby reducing the amount of fixed charges on which a return must be earned, and in consequence permitting the use of more coal-saving equipment.

In localities where the cost of fuel is high and little, if any, hydro-electric power is available, conditions may be entirely different from those where fuel is cheap. It would seem to be reasonably certain, however, that, if published records are correct as to the efficiency of certain recently completed moderate-pressure power houses, the burden of proof is on the high-pressure plant rather than on the moderate-pressure plant.

After all, simplicity of plan makes for constancy of operation and constancy of operation means better service, and that is one of the things we are paid for. It isn't the low cost of operation that counts—it is the high cost of shutdowns.

It still remains to be seen whether the high-pressure plant can operate as continuously and as economically a long period of years as the moderate-pressure plant.

Special Metals Required

for Extreme Temperatures and Pressures

By A. E. White and C. L. Clark

Respectively, Director and Associate
Department of Engineering Research, University of Michigan, Ann Arbor, Michigan

OPERATING temperatures of certain industries have increased to such an extent during the last few years that it is becoming a problem of great importance to secure metals capable of rendering suitable service under the conditions demanded. It is relatively easy to obtain metals suitable for service at temperatures up to 750 deg. F., more difficult to secure ones possessing necessary strength up to 900 deg. F., and if it is desired to use temperatures above this, the field of eligible metals becomes rather narrow.

The field as a whole may be divided into ferrous and non-ferrous divisions, with the first of these groups further divided into plain carbon and alloy steels, and the second into pure metals and alloys. In addition, another group which is becoming of more importance should be considered. This may be referred to as a "special surface" group and includes plating with such metals as chromium, nickel, etc., and nitriding.

Plain carbon steels find wide use in service at elevated temperatures not exceeding 750 deg. F. The properties of a 0.13 per cent carbon steel as determined by short-time tensile tests at various temperatures are shown in Fig. 1. It will be observed that the tensile strength decreases rapidly as the temperature is increased above

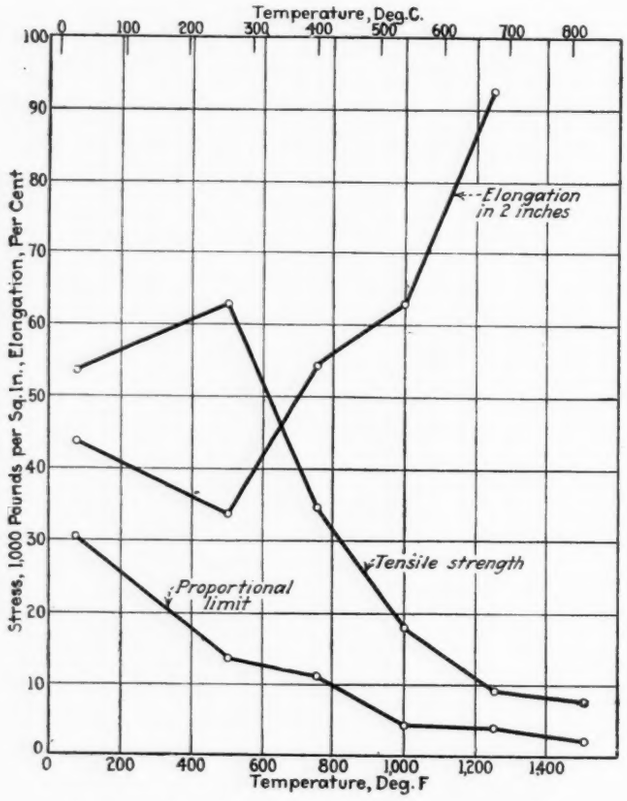


Fig. 1—Properties of 0.13 carbon steel at elevated temperatures as determined by short-time tensile tests

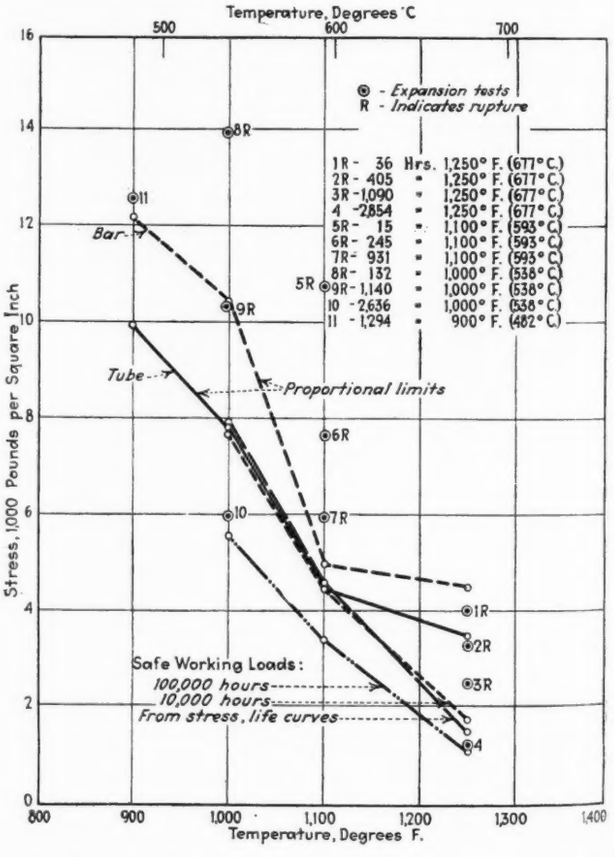

Temperature, Degrees C
500
600
700
⊙ - Expansion tests
R - Indicates rupture
1R - 36 Hrs. 1,250° F. (677° C.)
2R - 405 " 1,250° F. (677° C.)
3R - 1,090 " 1,250° F. (677° C.)
4 - 2854 " 1,250° F. (677° C.)
5R - 15 " 1,100° F. (593° C.)
6R - 245 " 1,100° F. (593° C.)
7R - 931 " 1,100° F. (593° C.)
8R - 132 " 1,000° F. (538° C.)
9R - 1,140 " 1,000° F. (538° C.)
10 - 2,636 " 1,000° F. (538° C.)
11 - 1,294 " 900° F. (482° C.)
Bar
Tube
Proportional limits
Safe Working Loads:
100,000 hours
10,000 hours
From stress, life curves
Stress, 1,000 Pounds per Square Inch
800
900
1,000
1,100
1,200
1,300
1,400
Temperature, Degrees F.

Fig. 2—Expansion results, safe working loads and proportioned limit valves for 0.38 carbon steel tubing

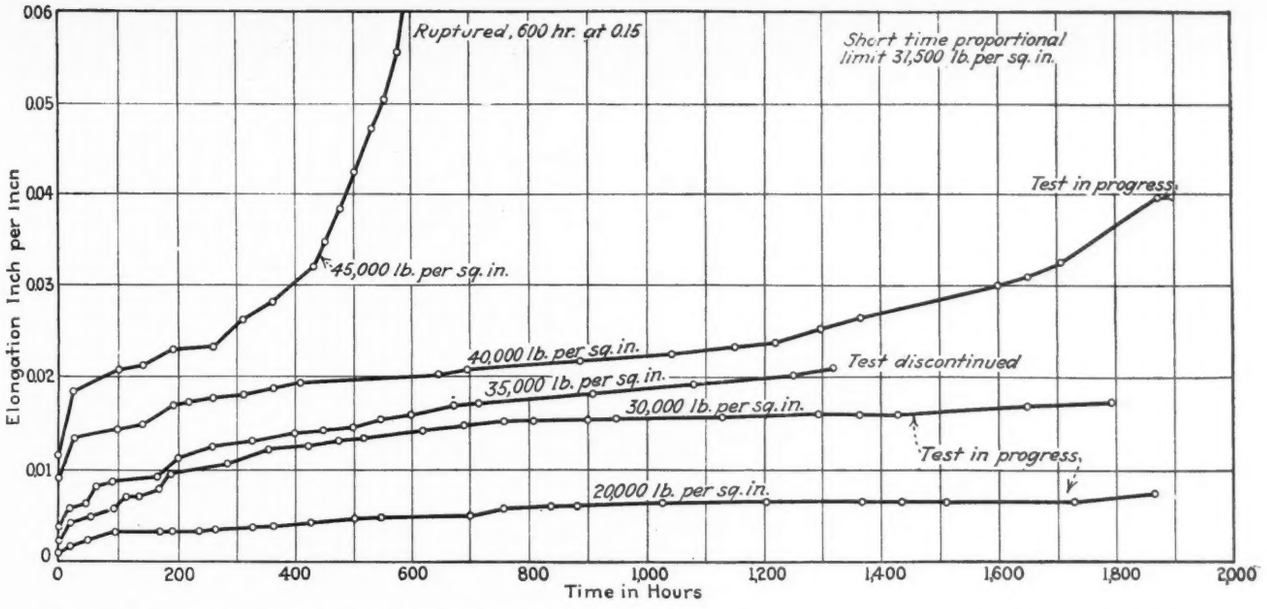


Fig. 3—Flow tests on 0.60 per cent carbon, 1.43 per cent chromium, 2.19 per cent tungsten steel at 1,000 deg. F.

500 deg. F., while the same is true of the proportional limit value at temperatures above 750 deg. F.

On the basis of results obtained from expansion tests on tube stock, the authors computed safe working loads for various periods of time for 0.38 per cent carbon steel. These values, together with short-time proportional-limit values at the corresponding temperatures, are shown in Fig. 2. It is seen that the stress producing rupture at 1,000 deg. F. in 100,000 hours, is computed as about 5,600 lb. per sq.in.; at 1,100 deg. F. as 3,400 lb., and at 1,250 deg. F. as 1,200 pounds.

As a result of these and similar tests, it becomes evident that for service at temperatures in the neighborhood of 1,000 deg. F., or above, alloy steels, rather than plain carbon ones, must be used. In the authors' opinion, the alloying elements of primary importance in increasing the load-carrying ability of metals in this temperature range are chromium, tungsten, and all other carbide-forming elements; while such elements as nickel, silicon and manganese, which are largely dissolved in the ferrite, are only of secondary importance in this temperature range, at least.

Chromium steels containing relatively large amounts of chromium are in rather wide use at present in high-temperature service. Their use is more or less limited by their cost, and attempts are continually being made to develop alloys possessing suitable properties for high-temperature work yet containing only small amounts of alloying additions.

The alloys that appear to offer the most promise at this time are those containing low percentages of chromium and tungsten. By means of suitable heat treatments, steels of these compositions will give proportional-limit values at 1,000 deg. F., of 40,000 lb. per sq.in. That they are also able to withstand large stresses for considerable periods of time is shown in Fig. 3.

The maximum short-time proportional-limit values obtainable with certain steels after being subjected to a large number of heat treatments are given in Fig. 4. This again shows the chromium-tungsten steels to be the superior.

Non-ferrous metals and alloys, although having certain fields of application in high-temperature work, are used to a limited extent at medium-elevated temperatures—temperatures up to 1,000 deg. F., on account of their cost. For temperatures above 1,250 deg. F., certain metals of this type, especially the nickel-chromium alloys, find wide application.

Certain of the "special surface" group of alloys are finding wide use through elevated-temperature service, but the group as a whole is not fulfilling all the hopes that were held for it. The use of metals and alloys plated with such metals as chromium or nickel has not yielded as satisfactory results as was expected. This is largely due to the differences in rate of expansion of the plated layer and the original metal, causing the plate to crack and peel off. The use of nitrided metals is meeting with more success, as here the nitrided layer is an integral part of the metal itself. Neither plating nor nitriding directly increases the load-carrying ability of a metal, however. Short-time tensile tests conducted by the authors on a chrome-aluminum steel at 1,000 deg. F. (538 deg. C.), both in the nitrided and un-nitrided conditions, showed the proportional limits to be the same.

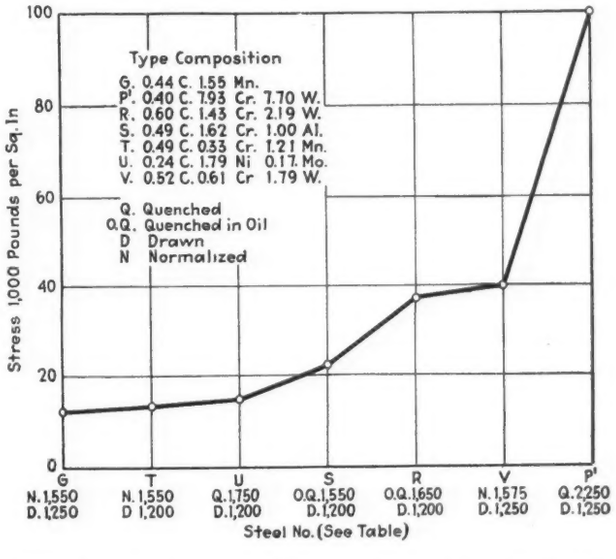

Type Composition
G. 0.44 C. 1.55 Mn.
P'. 0.40 C. 7.93 Cr. 7.70 W.
R. 0.60 C. 1.43 Cr. 2.19 W.
S. 0.49 C. 1.62 Cr. 1.00 Al.
T. 0.49 C. 0.33 Cr. 1.21 Mn.
U. 0.24 C. 1.79 Ni 0.17 Mo.
V. 0.52 C. 0.61 Cr 1.79 W.
Q. Quenched
Q.Q. Quenched in Oil
D Drawn
N Normalized
Stress 1,000 Pounds per Sq. In
100
80
60
40
20
0
G N. 1,550 D. 1,250
T N. 1,550 D 1,200
U Q. 1,750 D. 1,200
S Q.Q. 1,550 D. 1,200
R Q.Q. 1,650 D. 1,200
V N. 1,575 D. 1,250
P' Q. 2,250 D. 1,250
Steel No. (See Table)

Fig. 4—Maximum proportional limit values obtained at 1,000 deg. F. with various selected steels

Modern Developments Demand *Better Metals*

By R. J. S. Pigott
Consulting Engineer, Stevens & Wood, Inc., New York City

DURING the period when 200 lb. and 550 deg. was the "high" range for steam plants, there appeared little or no demand for special materials. The service life and utility of cast iron, mild steel and bronze was sufficiently great so that no special search had to be made for metals suitable for severe service. But the advent of the higher pressures, from 400 to 1,800 lb. and especially the introduction of higher superheats, 750 deg. for steam and 1,000 deg. for certain industrial processes, has greatly emphasized weaknesses in the earlier simple metals or alloys that were so small under the lower pressures and temperatures as to be practically indistinguishable.

We are interested in the strength (particularly the elastic limit) and the shock resistance of metals in all pressure vessels, such as boilers, piping, feed-water heaters, turbine casings and pumps. The same interest applies in all rotative equipment, such as turbine rotors, generators and motors, pump and fan impellers. Corrosion resistance is of moment in all the foregoing, while resistance to erosion is especially important in turbine blading, valve seats, pump and fan impellers.

> Contrary to the general opinion, available metals are satisfactory for the present-day high-pressure boiler. It is in the construction of other elements, as Mr. Pigott clearly brings out in this article, that better metals must be employed if full advantage is to be taken of the modern trends in power-plant design.

So far as the boiler is concerned, we are likely to use present steels for some time to come. The nature of manufacture, drilling, shearing, bending and calking for work up to 800 or 900 lb., coupled with forging for 1,200 lb. and higher, largely limits us to ordinary steel; many of the high-strength alloys are deficient in ductility, and many likewise will not stand calking or welding. The high-strength alloys, with few exceptions, have an advantage in strength only at the lower temperatures; at temperatures of 900 deg. and above, all the steels come down much to the same value in strength.

In practically all cases the alloy steels cost so much more than regular boiler steel that the slight possible gain in strength or other qualities is completely overshadowed by the enormous increase in cost. The price range may be as much as fifty cents or one dollar per pound for alloy steel against two or three cents for boiler plate.

Tubes in most cases have not yet been successfully drawn in alloy steels. Some progress has been made in welded stainless iron tubes, but commercially the product is hardly satisfactory yet. The inside of a tube in a superheater may be at a temperature of 750 to 800 deg.—at the same time the outer surface is up to 1,100 or 1,200 deg. Under these circumstances no alloy steel shows any particular advantage in strength over ordinary tube steel; it has more resistance to corrosion and oxidation, but the primary need is for more strength.

On the other hand, ordinary seamless steel tubes up to $\frac{3}{8}$ in. thick are working successfully in oil refineries where the inside conditions are 900 lb., 1,000 deg., and the outside fibers are without doubt as high as 1,400 deg. Alloy steels would not help this situation materially even if the price were low enough. There has been a limited use of stainless iron tubes for tar-stills, where ordinary steel tubes are attacked by water and sulphur and brass tubes by ammonia and sulphur. When the difficulty with cracking the stainless iron tubes by rolling-in is overcome, this use will grow.

Autogenous welding of pressure vessels is bound to increase; we have used gas and arc welds for years in piping and are beginning to approach the welding of pressure vessels with less fright. For a considerable time yet the only welding for these purposes will be on low-carbon steel, as the welding of alloy steels containing chromium is not yet on a sufficiently established basis.

Soot blowers located within five or six rows of tubes from the fire have always given trouble, both from oxidation and warping. The use of calorized tubes (aluminum-coated) has been attended by moderate success where the conditions are not too severe. The later alloys are nickel-chromium compounds, such as Michrome and Dialloy; these resist corrosion well, but since they soften at relatively little higher temperature than steel, there is still some trouble from warping. These alloys, however, have sufficiently better qualifications for the service to yield a noticeable decrease in trouble from soot blowers in the first pass of the boiler.

For piping and valves there is no demand for alloy steels from the point of view of strength. Ordinary lap-welded and seamless drawn tubes have satisfied all the requirements. There is some tendency to use seamless tubing for smaller piping at 400 lb., and quite generally for higher pressures; but the writer cannot see any

special necessity for ruling out welded piping for any pressure. If the seamless tube can be bought for about the same price as welded pipe, it should be used.

The main interest in alloys is for parts of valves, such as seats, disks and stems, subject to corrosion and erosion. The problem in seating materials is fivefold:

(*a*) Resistance to corrosion, from salts in steam or water, and from included air or CO_2.

(*b*) Resistance to erosion, especially for the flow of wet or dirty steam.

(*c*) Good seating-metal qualities, to avoid scuffing or galling when the seat and disk ride over each other.

(*d*) Maintained high strength at high temperature.

(*e*) Avoidance of distortion, to maintain good seat contact.

The older bronzes gave satisfactory strength and resistance to corrosion for temperatures up to 550 deg., especially the nickel-bronzes, but show high erosion, and low strength for higher temperatures. Monel metal had for a number of years nearly a monopoly for high-temperature work; but while it has good strength at high temperature and excellent resistance to corrosion, it is not satisfactory in resistance to erosion, and it galls very badly. Monel is always a difficult shop material—very difficult to get sound castings of any but simple shapes, and difficult to cut in both cast and forged or rolled form. One other copper-nickel alloy, the Hopkinson "Platnam" used in England, eliminated the galling by the use of tin (13 per cent) but in other respects was as bad as Monel—hard to cast, hard to cut, and further limited for high-temperature use by the presence of the tin.

The search for better materials took a long time, and these entered the market slowly, on account of the wide use and trust in Monel. The stainless steels and irons, and to a small extent the high nickel chromium steels, are rapidly taking the place of Monel for valve seats, disks and stems. Acceleration tests on seats and disks set 0.003 in. apart and blown with 250 lb. steam for about 350 hours showed the resistance to erosion about as follows:

1. Nickel chromium steel (20 per cent Ni.; 75 per cent Cr.).
2. Stainless Steel (14 per cent Cr.).
3. Monel and forged Everbrite (30 per cent Ni.).
4. Cast Nickel-Copper alloys (25 to 30 per cent Ni.).
5. High tin bronze (12 per cent Tin).

The first two were very noticeably better in resistance to erosion than the others.

Another quality required for seating materials is close grain. It follows, then, both in theory and experimental proof, that forged materials will always be better than cast, in the same alloy. They are denser, more homogeneous and finer-grained. When it is realized that the actual seat of a safety valve is from $\frac{1}{64}$ to $\frac{1}{32}$ in. wide, and the valve is seated with relatively little pressure, the importance of resistance to erosion and close grain is very evident.

None of the modern materials are as yet good enough in bearing qualities—they will all gall, although not as much as Monel. There is promise in the new "nitrided" steels, in which an extremely hard casehardening is produced without distortion. There is also in chromium plating a possibility that has not been exploited yet.

It should not be forgotten that distortion will have as much to do with seat tightness and durability as qualities of the metal itself. Distortion is a function almost entirely of valve design, complicated by the fact that all the present materials, Monel, stainless steel, and nickel-chromium alloys, have coefficients of expansion exceeding those of cast or forged low-carbon steel by 25 to 45 per cent. Consequently, distortion is bound to occur when a valve that is assembled cold is heated to 600 or 700 deg. F., aside from the additional distortion that may come from pressure effects on the valve body. There are two ways of avoidance:

(*a*) Make the seat in such a way that body distortion does not reach it.

(*b*) Make the body of a material of the same coefficient of expansion as the seat.

Turbine blades, likewise fan and pump impellers, have to withstand not only corrosion and erosion, but also centrifugal and load stresses. Monel is disappearing in turbine blading, to be replaced by stainless steel in most cases, and a high-nickel chromium steel in the case of one large company. These steels are strong enough for all but the last-stage wheels of large turbines. Here nickel steel is used, as the mechanical stresses are very high, and resistance to erosion must be sacrificed to some extent, to get a higher strength.

In the case of fan impellers, especially those handling coal dust or flue dust, several alloy steels, including manganese steel, have been used, including rubber and lead coatings; but the net result is that ordinary low-carbon steel is cheap enough to overbalance the extra maintenance of the high-priced alloy steels, and continues in general use.

The stresses in pump impellers are relatively low; corrosion and erosion are sufficiently reduced by bronze; for sulphite and similar work aluminum bronze, manganese bronze and ordinary phosphor bronze (90 : 10) are often used; while Monel and lead-coated impellers are reserved for chemical work.

Bolting, for high-pressure, high-temperature steam, has definitely moved into the alloy group such as Simplex, Mayari and other steels of a strength up to 125,000 lb. per sq.in. These steels gave trouble at first from extremely variable mechanical properties, tensile strength varying from 80,000 to 130,000 in the same lots of presumably similar steel, and ductility likewise, but generally low. After three or four years of battle between makers and users these materials have settled down, like the specifications, to more reasonable variations. Part of the trouble was ignorance of steel displayed in the specifications, and part was poor control of manufacture.

It has been hard to find suitable spring steels for high-temperature safety valves and similar work. Ordinary carbon steels will not stand 600 deg. F. without set, under the spring stresses used in safety valves. But most of the manufacturers now use a tungsten steel claimed to be good for 900 deg., actually about the safe limit for permanence is around 750 deg. Fortunately, the outside spring of a safety does not have to stand the full steam temperature, but only the conduction heat, and a short-time blow of leakage steam past the spring.

To summarize, we are in much the same condition as always—a neck-and-neck race between severe service conditions and metals that will meet them at commercial prices. We have relatively new steels for turbine blading, but still no one steel covers the field, and none has much margin. The same is true of seating materials; we still need a high-temperature metal that will stand erosion and corrosion and still avoid galling.

Metals Unaffected by Embrittlement Needed

By Frederick G. Straub

Special Research Assistant in Chemical Engineering, University of Illinois, Urbana, Ill.

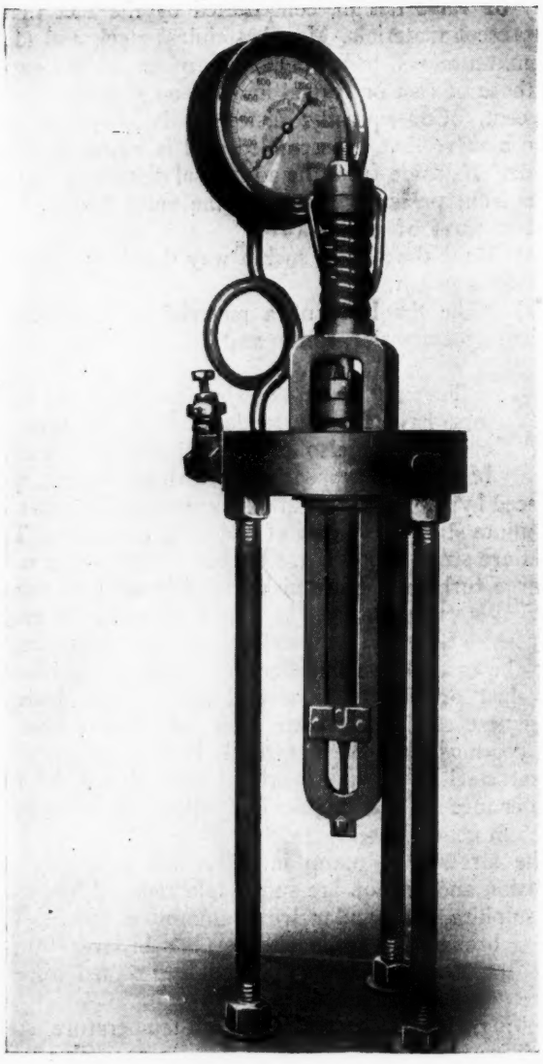


Fig. 1—Tension section of apparatus used in embrittlement tests

THE action of caustic soda solution on steel under tension to cause embrittlement has been recognized for some time. This action has been causing enormous losses to the operators of steam boilers. Apparently, the increase in use of higher steam pressures has caused an increase in the occurrence of these losses where special precautions to prevent its occurrence have not been taken. Consequently, the subject of embrittlement is of paramount importance to the manufacturers as well as the operators of all steam boilers, particularly the higher-pressure boilers.

The research work conducted at the Engineering Experiment Station, University of Illinois, in co-operation with the Utilities Research Commission of Chicago, has done much toward determining the actual cause of this type of cracking and methods to be used to prevent its occurrence. The cracking is due to the action of the caustic soda in boiler water concentrating in the seams, where it attacks the highly stressed metal adjacent to the rivets. Some instances are on record where boiler tubes have failed in this same manner, but the cracking predominates in the riveted areas. The presence of certain salts along with the caustic apparently stops this cracking. Consequently, some relief is obtained by the use of properly supervised water treatment. Methods of water treatment to accomplish this purpose are described elsewhere[1].

By inside calking of all seams it is hoped to prevent the seepage of the boiler water into the riveted area, and if seepage does take place, the absence of outside calking allows free leakage and serves to indicate that the caustic is entering the seam. Much hope is being placed in the effectiveness of this method of manufacture. Seamless drums are being made for higher pressures, which, of course, preclude the possibility of seam trouble, but even these drums still leave some possibility of cracking taking place in the tubes.

Much effort has been spent in trying to develop a type of steel or alloy suitable for boiler use which will not be attacked by the caustic. If this could be accomplished, water treatment could be simplified very much and the danger of embrittlement eliminated.

Figs. 1 and 2 show the test equipment that has been used to run tests on the various steels. The steel is under a definite tension produced by the compression of the spring. Any predetermined load may be maintained on the specimen at all times. The specimen is also in contact with the caustic solution at a temperature corresponding to the steam pressure maintained. The majority of the tests have been made with a steam pressure of 500 lb. gage. Tables I and II give the chemical analyses and physical properties of the steels

*This article is an elaboration of part of the data appearing in Bulletin 177 of the Engineering Experiment Station, University of Illinois, entitled "Embrittlement of Boiler Plate," by Samuel W. Parr and Frederick G. Straub.

[1]"Water Treatment to Prevent Embrittlement," by F. G. Straub, paper read before the American Water Works Association, June 12, 1928.

TABLE I—CHEMICAL ANALYSES OF METALS TESTED

Results in per cent composition

Reference	Carbon	Manganese	Phosphorus	Sulphur	Silicon	Nickel	Vanadium	Copper	Molybdenum
F.S.	0.18	0.45	0.012	0.027	....	....	....	....	....
Ni	0.18	0.54	0.035	0.028	0.23	2.62	....	....	....
G.	0.06	0.33	0.012	0.035	....	....	....	....	....
I	0.17	0.41	0.020	0.031	0.05	0.03	....	....	....
A	0.42	0.46	0.036	0.044	0.05	....	....	....	....
1112	0.15	0.65	0.086	0.156	0.03	....	....	....	....
Mn	0.35	2.13	0.020	0.026	0.02	....	....	....	....
M. I.	0.02	0.017	0.003	0.010	0.01	....	....	....	....
Si....	0.34	0.75	0 027	0 034	0 29	....	....	....	....
CA	0 05	0 31	0 010	0 019	0 01	....	....	0 46	0 12
V	0 03	0 09	0 006	0 036	0 06	....	0 24	....	....

TABLE II—TENSION TESTS OF METALS TESTED

Reference	Yield Point	Ultimate Tensile Strength	Production in Area, Per Cent
	Pounds per Square Inch		
F.S.	35,200	60,400	61.0
Ni	45,000	75,300	59.3
G.	32,000	56,500	65.3
I	32,000	56,500	67.5
A	33,000	74,500	46.0
1112	41,000	63,500	58.0
Mn	4,,000	103,000	55.0
M.I.	30,000	50,700	72.0
Si	43,000	87,000	48.0
C.A.A.	45,000	55,000	73.0
V.	47,000	59,000	73.0

TABLE III—EFFECT OF CHEMICAL COMPOSITION ON RATE OF EMBRITTLEMENT

Steam pressure 500 lb. gage. Concentration of solution 300 grams sodium hydroxide per liter.

Reference	Yield Point	Lowest Load at Which Cracking Takes Place, Lb. per Sq.In.	Rate of of Cracking F. S. as Unity
F.S.	35,200	35,500	1.0
Ni	45,000	47,000	1.0
G.	32,000	30,000	2.5
I.	32,000	44,000	1.0
A.	33,000	30,000	2.5
1112	41,000	45,000	0.25
Mn	47,000	50,000	2.0
M.I.	30,000	30,000	1.0
Si	43,000	55,000	4.5
C.A.A.	45,000	39,000	1.0
V.	47,000	40,000	1.0

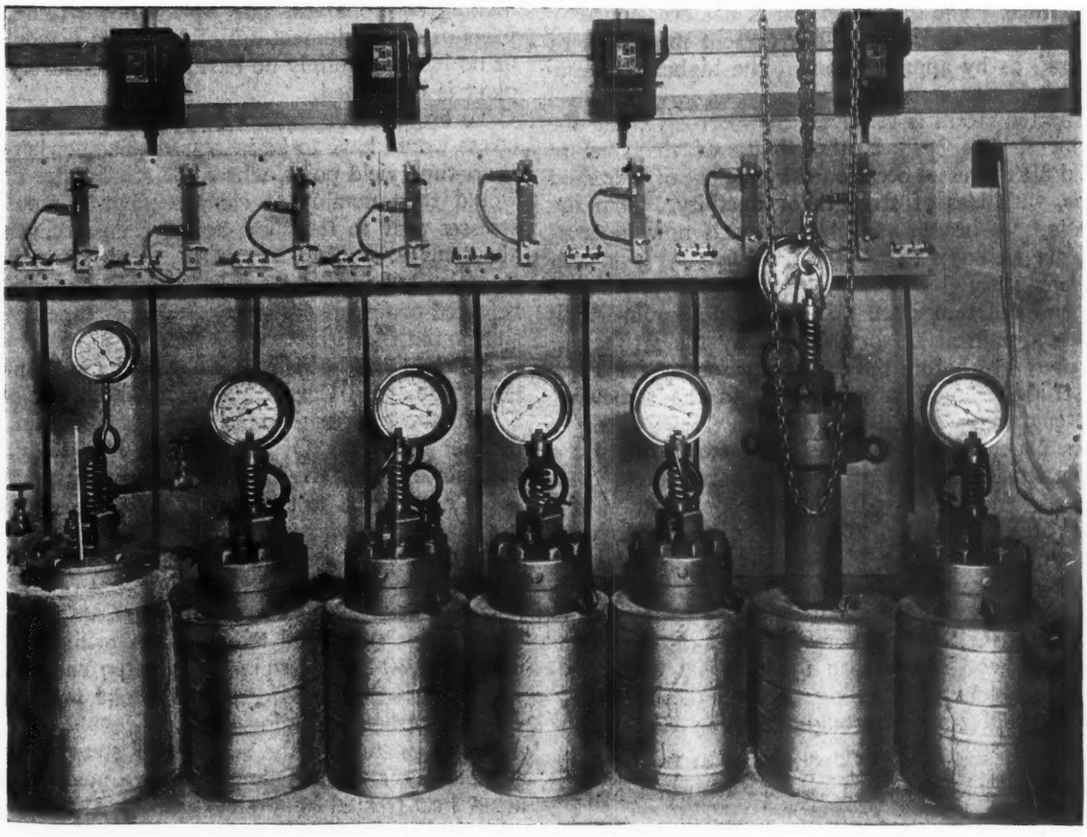
and alloys tested. Table III gives the results of the tests run to determine the ability of the metals to withstand the caustic attack. A large number of tests have shown that the caustic apparently penetrates normal boiler steel at a definite rate (time rate of penetration)[2] when the concentration of caustic and steam pressure are constant. In the tests reported the rate of caustic penetration of normal boiler steel is taken as unity and the rates of the metals tested compared on this basis.

Steels with practically all the constituents varied through wide limits have been tested, as well as a few steels alloyed with nickel, vanadium, copper and molybdenum. As shown in the results given in Table III, one steel has a slower cracking rate than ordinary boiler plate. This one was a sample of screw stock high in sulphur and phosphorus. These results indicate that if a steel or alloy is to be developed that will withstand the caustic attack, it will undoubtedly be some special alloy, since reduction of impurities in steel or slight changes in chemical composition have but small effect upon its resistance to caustic attack.

[*In addition to the reference on methods of water treatment that will inhibit caustic embrittlement, given in the footnote on page 902, information on this topic is given in the article by R. E. Hall, on pages 873 to 875 of this number of "Power." Doctor Hall recommends the addition of chemicals to the boiler water that establish and maintain a protective film on the surfaces subjected to the corrosive action of a high alkaline concentration. A phosphate film is recommended, which prevents the caustic from coming into contact with the metal.*—EDITOR.]

[2]Bulletin 177, Engineering Experiment Station, University of Illinois, page 36.

Fig. 2—Test units used for high-pressure tests



Steels Are Now Available

for High Pressures and Temperatures

By L. W. Spring and J. J. Kanter

Respectively, Chief Chemist and Metallurgist, and Testing Engineer
Crane Company, Chicago, Ill.

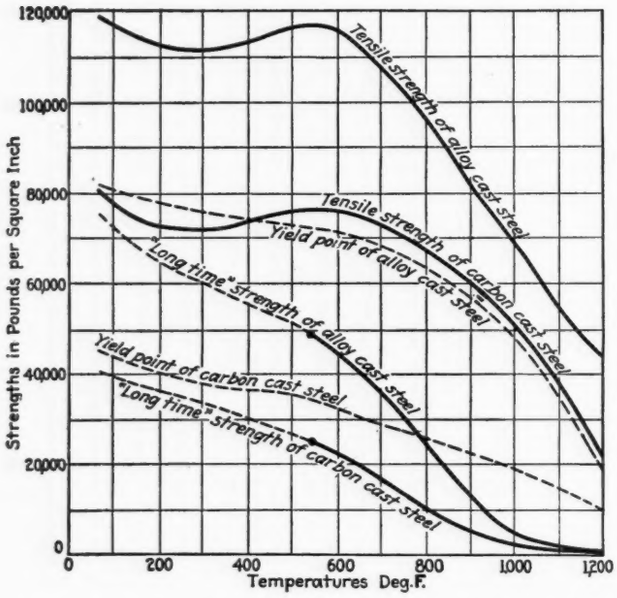


Fig. 1—Comparison of high-temperature strength of carbon and alloy cast steels

The "long-time" strength curves show the limiting stresses producing not over one per cent of flow in a year, the dotted portion of the curve corresponding to the proportional limit.

FOR the last few years power-plant engineers have been concerning themselves with the stability of metals at temperatures encountered in modern steam practice. The great concern is not occasioned so much by actual difficulties at the present maximum of 750 deg. F. as by apprehension of the higher temperatures inevitably to come.

Superheated steam is relatively non-corrosive, and the chief concern has been in the ability of the steel to withstand fiber stresses occasioned by the use of increased pressures, with loss of strength or ductility. That the high-grade open-hearth and electric steels now in common use are well able to meet such service has been well substantiated through careful long-time "flow" or "creep" tests at elevated temperatures. This information, with much data discussed at some length by the writers in a paper presented before the American Society for Testing Materials,[1] should be reassuring to the somewhat troubled engineer.

In this paper it was reported that steels that have undergone small amounts of flow do not seem to have deteriorated, tensile properties, including elongation and reduction of area, remaining approximately what they were before the flow tests. Since little or no deterioration occurs or should be expected (unless through oxidation), and since with ductile metals at temperatures above 600 deg. F., 10 or 15 per cent or more of flow or elongation is necessary before the material starts out on its final stretching preceding rupture, therefore any slight flow or deformation brought about by stresses below the proportional limit should not be seriously objectionable unless interference with operation results.

Usually, the factors of safety employed are sufficient, but should stresses be reached which produce flow, there is a remote probability of rupture with danger to life or property until at least 10 to 15 per cent of total flow or distortion has taken place. That is, should proper factors of safety be neglected and a tank, pipe, valve or other apparatus be subjected continuously to high enough temperature and sufficient pressure be built up to produce even circumferential flow or stretch of 0.01 in. per inch per year, it would result simply in the slight expansion of the tank, pipe or valve. Unless such increase of size interferes with operation or serviceability by making the apparatus take up too much room, throwing parts of the equipment out of line, etc., it would do little or no harm, certainly no harm so far as safety to life or plant is concerned.

To bring about rupture in a short period of time, the stress would have to be increased to somewhere near the short-time yield point, where the steel under observation would start notably to elongate; but it would require stresses near to the short-time tensile strength to bring about rupture within any comparatively short time. Continuation of some of the tests reported upon in the paper referred to for much longer periods of time have further substantiated the foregoing observation.

In Fig. 2 is shown a creep test of ten months' duration made at 800 deg. F. under 10,000 lb. per sq.in. load. The total creep for the ten months is 0.0009 in. per inch, or 0.09 per cent. Of this amount 0.00015 in. per inch took place during the last four months of the test, or, in other words, the rate of creep progressively decreased. However, if creep were to continue at the average rate indicated by the last four months of the test, over 21 years would be required to produce but one per cent of flow or deformation. Actual fiber stresses used in design for high pressure and temperatures ordinarily are well below 10,000 lb. per sq.in. The creep curves in Fig. 2 show the stresses giving not over one per cent a year for cast carbon and alloy steels.

In anticipation of the advance of steam temperatures and pressures the metallurgist is in a position to recommend heat-treated alloy steel castings should something

[1] "Long-time" or "Flow Tests of Carbon Steels at Various Temperatures With Particular Reference to Stresses Below the Proportional Limits."—Proc. A.S.T.M. 1928, also *Valve World*, August, 1928.

better than the plain carbon steels be required. Alloy steel castings have been used for several years in the oil refinery and now stand available as a development to meet the future needs for power plants. Fig. 1 shows superiority of nickel chromium alloy cast steel in both short- and long-time tensile tests. At atmospheric temperatures it is seen that alloy cast steel has 70,000 to 80,000 lb. per sq.in. yield point, against 45,000 lb. per sq.in. for cast carbon steel. The strength advantage is retained throughout the temperature range.

As a further example, not so many years ago an improvement in bolting material became necessary in order to cope with the stresses on joints brought about by increased pressures. The difficulty was quite effectively solved by substituting comparatively inexpensive heat-treated alloy steels for the wrought and rolled carbon steel then in vogue. Although a number of compositions are in use, the one most often specified is of the nickel chromium series containing 1.25 per cent nickel, 0.75 per cent chromium, 0.40 per cent carbon (S.A.E. 3140). This steel usually is heat-treated to give over 100,000 lb. per sq.in. elastic limit. It is tough, sufficiently ductile and retains sufficient strength at high operating temperatures to insure an ample factor of safety.

The chief use that steam practice has made of corrosion-resistant alloys thus far is for turbine blading, superheater tubes, valve trim and the like. The primary qualifications for a blade material are resistance to pitting, corrosion and erosion as well as proper strength. This has led to the adoption of stainless iron containing 11 to 13½ per cent chromium and less than 0.12 per cent carbon.

This alloy has excellent strength up to 800 deg. F., but at higher temperatures becomes somewhat less stable. It has been realized for some time that for superheater tubes where red heats must be withstood, a material superior to the carbon or ordinary chromium stainless steels must be used. Even stainless steels of 18 per cent chromium content have proved unsatisfactory for tubes because of an embrittling effect at these high temperatures manifested both in superheater and oil-cracking still service.

Where temperatures in the neighborhood of 1,000 deg. F. and above are used, there now is widespread adoption of the non-magnetic, austenitic, nickel-chromium steels. The steels of this series contain anywhere from 8 to 60 per cent of nickel, 8 to 20 per cent of chromium and usually not over 0.2 per cent carbon. These alloys are comparatively soft, about 200 Brinell; of somewhere around 100,000 lb. per sq.in. tensile strength; 40,000 to 50,000 lb. per sq.in. yield point; 40 to 50 per cent elongation in 2 inches; and respond only feebly to heat-treatment. It may be said that all these austenitic, nickel-chrome steels constitute a decidedly superior class of metals for strength, scaling resistance, or in general, mechanical and chemical stability at red heats. An alloy of this series now being widely adopted for tubes by some power plants as well as oil refineries contains 8 to 9 per cent of nickel, 18 to 20 per cent chromium and 0.5 to 0.15 per cent of carbon. Aside from having excellent resistance to oxidation, long-time creep tests show this alloy capable of withstanding 3,000 lb. per sq.in. fiber stress at 1,400 deg. F. with no appreciable deformation during tests that lasted considerable lengths of time.

There seems to have been a prevalent fear that all heat and corrosion resistant alloys are liable to embrittlement upon exposure for any considerable time to red heats. This feeling possibly has been brought about by the common knowledge that plain 18 per cent chrome steel has such a shortcoming. However, most other types of resistant steels, fortunately, do not seem to be thus affected. Pronounced embrittlement appears to be quite unique to the plain high chrome steel, no such action of any consequence having been observed even for the lower chromium stainless irons and steels (11 to 13½ per cent chromium). The non-magnetic nickel-chromium steels, being quite stable structurally at all temperatures, are not prone to lose much of their natural great toughness, shock resistance and ductility under conditions that cause the 18 per cent alloy to become of but little use for high-temperature work.

With regard to some of the theoretical aspects of the special alloy steels that have been introduced for high-temperature service, it is generally understood that they are divided into two general classifications—those which respond to heat-treatment and have critical transformations, and those which are non-magnetic or only feebly responsive to heat-treatment. The former are often designated as "alpha" or sometimes as "martensitic" on account of their metallographic and crystal structures. Similarly, the high-nickel, high-chromium iron alloys are known as "austenitic." At the present time there seems to be some pessimism over the suitability of anything of the "alpha" nature as an alloy for very high-temperature service (over 1,000 deg. F.). Medium chrome-medium tungsten steel, a material in this class, is considered by some to be promising for red heat, but that remains to be proved. Eventually, for applications at very high temperature it does seem that economical service perhaps will be best afforded by some of the more or less commercially available (though comparatively difficult machining) steels of the "austenitic" type as these tend to keep their good qualities.

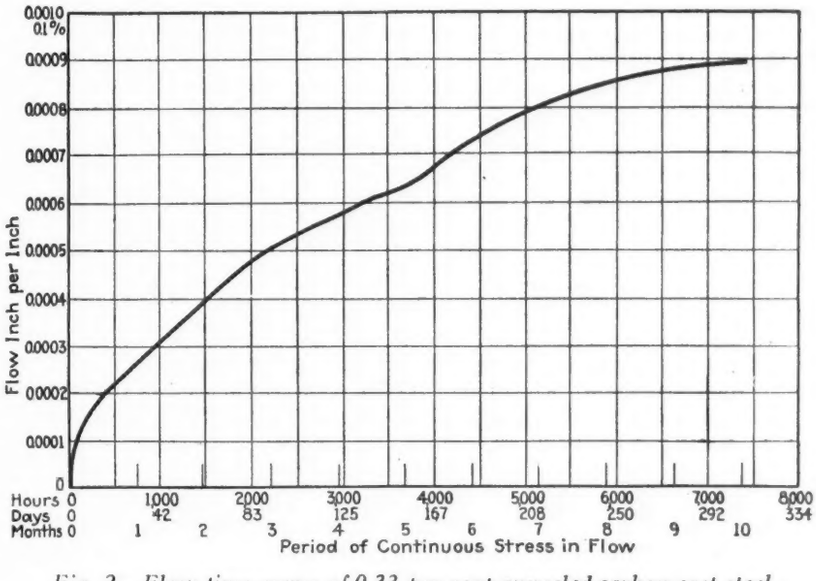


Fig. 2—Flow-time curve of ***0.33 per cent annealed*** *carbon cast steel tested at* 800 *deg. F. under load of* 10,000 *lb. per sq.in.*

What Pressure Pays Best?

A SYMPOSIUM OF ENGINEERING OPINION

By Sabin Crocker
Engineer, The Detroit Edison Company

No general agreement appears to exist as to the best steam condition for central station plants, but the trend to pressures over 1,000 lb. is clearly evident from the opinions expressed in this symposium

IN THE past decade much attention has concentrated on higher steam pressures and temperatures for central stations and industrial plants. Several valuable papers on various phases of the subject have been presented before the A.S.M.E., and numerous articles have appeared in the technical press from time to time. The early papers dealt principally with the theoretical possibilities of various ideal cycles, which were later supported with more or less exact estimates of relative cost and economic worth. Finally, when plants embodying these advanced features were actually built, it was shown that the gains predicted from an ideal cycle basis were sometimes exceeded in practice through secondary factors such as reduced moisture content in the lower turbine stages, better combustion with preheated air, and the like. The apparently visionary ideas of 1922 and 1923 have become the accepted facts of 1929 plant design. The present article is not written with the thought of presenting original design features, but is, rather a summary of how others have demonstrated the courage of their convictions by building plants embodying their ideas.

Apparently, no general agreement exists as to optimum steam conditions or heat cycles as measured by the economic yardstick. This is due, in some instances, to the fact that the new cycles have been superimposed on existing plant equipment with the intention of getting the most out of building space and equipment already available. Under some circumstances the addition of high-pressure equipment and reheat in an existing plant involves little extra expense for buildings and improvements to land, and hence affords increased capacity at relatively low capital cost, with decided betterment to station economy. The first 1,200-lb units were installed under such circumstances, and apparently justify their designers' expectations.

The choice of steam conditions for the high-pressure section of such a station is determined within rather narrow limits by the requirements of the existing low-pressure station and by the temperature limitations of materials commercially available. A much wider choice of cycles is available to select from in starting the design of an entirely new station, and it is not surprising that there is a wide variety of opinion as to which cycle is most advantageous. The Committee on Power Generation of the Association of Edison Illuminating Companies attempted in 1927 to determine a consensus of opinion regarding the most economical steam pressure for power stations. The replies received from power companies that had studied the subject varied so radically that the committee deemed it advisable to postpone arriving at a consensus opinion and to confine its findings to statements of fact regarding which agreement could be attained.

Certain features of plant design are rather definitely established by physical limitations of material and equipment, and are quite generally accepted as existing. Among these features are:

1. The maximum amount of moisture permissible in the lower turbine stages without producing excessive erosion. American practice usually limits the moisture in steam entering the condenser to 10 or 12 per cent. With a throttle temperature of 700 to 750 deg., this moisture limitation precludes the use of throttle pressures exceeding about 500 lb. without resorting to reheat.

2. The maximum total temperature possible with carbon-steel pipe and fittings, superheater tubes, etc. The strength of carbon steel falls off so rapidly somewhat above 750 deg. that its use is not generally considered practicable above that temperature. The use of much higher temperatures awaits the development of suitable materials on a commercial scale.

3. The highest pressure practicable with riveted drum bent-tube boilers. The bent-tube boiler has a relatively low first cost for pressures below about 600 lb. At higher pressures design limitations make it necessary to use forged drums or to resort to the header type straight-tube boiler, both of which increase the cost.

An interesting sidelight on the previously mentioned lack of agreement regarding the optimum cycle for new plants is afforded in the following quotations from designers of recent plants. Credit to the source of these quotations has been omitted intentionally to avoid any

suggestion of personalities entering this presentation.

"In the final analysis in which investments and operating charges are balanced against thermal efficiency, it is apparent that a working pressure of 600 lb. offers the best return on the dollar at this time (1927)." This quotation refers to a 600-lb. reheating regenerating cycle with throttle and reheat temperatures of 700 degrees.

"Detailed estimates on the first cost of the station designed for 400 lb., no reheat, two-and-three-stage extraction; and 550 lb. reheat; and 1,200 lb. reheat, four-stage extraction, were prepared and from these estimates the fixed charges were determined. When considering the fixed charges plus the production cost, it appeared that the 550-lb. pressure reheat plant was less economical than the 400-lb. pressure no reheat plant, because the saving in B.t.u. consumption of the former was more than offset by the increase in fixed charges. The 550-lb. plant was also more expensive in both first cost and B.t.u. consumption than the 1,200-lb. plant. This study left but one conclusion, namely, that the station should be designed for either 400 lb. pressure or 1,200 lb. . . . The 400 lb. pressure, no reheat, three-stage extraction cycle is the most economical with coal below $3.50 per ton, and with coal above that price the 1,200 lb. pressure, reheat, four-stage extraction cycle is the most economical." This quotation refers to a study made in 1928.

"It was found that if 600 lb. was used, compared with 350 lb., a saving in ten years' time of one million dollars worth of fuel would be effected. A rather attractive fuel saving to be sure; but on going back to the fixed-charge tabulation, it was found that the summation of fixed charges over ten years for the 600-lb. plant would exceed by two million dollars the fixed charges on the 350-lb. plant. Therefore, by adopting the higher pressure $100,000 worth of fuel a year would be saved at the expense of $200,000 of the stockholder's funds. Naturally, the 350-lb. cycle was adopted for the station.

"This does not mean, however, that the 600- or 1,200-lb. plant does not have its place. The capacity use factor over the ten-year period was 56 per cent and the fuel cost was $2.35. Naturally, if the fuel cost had been $4.70, then the fuel saving would have been two million dollars, the additional fixed charge would have been two million dollars, and the two systems would have been on a par. Furthermore, if the load factor were increased from 56 per cent to some higher factor, then the two systems would have been more nearly equal. Similarly, a combination of a somewhat higher fuel cost and a somewhat higher load factor would tend also to effect equalization."

"It is considered that the adoption of 400 lb. pressure has been shown to be preferable on the score of simplicity, ease of operation and cost. Should conditions ever arise under which the use of a pressure of 1,200 or 1,500 lb. may become attractive, the proposed pressure will fit in nicely. The 600-lb. plant will not."

"In the past station economy has been improved by increased steam temperatures as well as by increased steam pressures. Steam temperatures have increased from 500 to 700 degrees. The temperature limit of 700 deg. has been chosen simply because materials are at hand that can safely be used under this total steam temperature. The only parts subject to the effect of high temperature are the superheater and its fittings, the steam piping, valves, drips, and the high-pressure section of the turbine. The remainder of the plant equipment would be that usually furnished for the pressure selected."

In this respect the high-temperature cycle is superior to the high-pressure cycle since it does not require a superior standard of construction for the boilers, boiler feed pumps or feed-water piping and does not use additional auxiliary power for driving high-pressure feed pumps. An outstanding advantage is that neither a reheat boiler nor a live steam reheater is required.

"If the B.t.u. per kilowatt-hour of the plant can be decreased by increasing the steam temperature above 700 deg., the boiler and condenser units will be decreased in size and hence in cost, while the only increase in cost will be for those parts already enumerated with which the highly superheated steam comes in contact. These parts form a small portion of the plant and their proportionate cost is not great. If such parts could be secured at a reasonable price, there are reasons to believe that a plant with high superheat and moderate pressure would cost less than a high-pressure plant for the same economy. With the higher steam temperatures a greater part of the expansion in the turbine would take place in the superheat region than with high-pressure plants, resulting in a more economical turbine."

"Considerable gains are possible by the use of high steam temperature at moderate pressures, and this should stimulate development of metals for high-temperature service."

Still greater gain in thermal economy can be obtained, of course, through the combined use of high temperature and high pressure. The foregoing reference is of particular interest in that it points out another way of obtaining a substantial gain in economy without having to pioneer in both fields, high pressure and high temperature, at the same time.

Some idea of the relative cost of 400-, 600- and 1,200-lb. stations can be obtained from the following statements of leading exponents of 600 and 1,200 lb. pressures. According to these quotations the additional capital investment required for a 1,200-lb. plant over a 400-lb. plant lies between $5 and $40 per kilowatt of installed capacity. The proponents of the lower figure appear to be in the majority, and support their case with the claim that the cost of 1,200 lb. equipment will come down as its use increases.

"Based on the estimates we find that the average cost per kilowatt for the first two 1,200-lb. 55,000-kw. units will be less than $105 a kilowatt, this including land, engineering, supervision, interest during construction, all overheads, outdoor substation, and everything necessary to complete the plant in its entirety." Based on 1928 estimates.

"Feed-water pumps and heaters are somewhat more expensive for the higher pressures. The major increase in cost of piping and valves for 1,200 lb. over 400 lb. is due to the added reheat piping and the increased cost of boiler-feed piping. The steam piping for higher steam pressures costs very little more on account of its reduced size."

"The total effect of these items results in an excess cost of the 1,200-lb. station (over the 400-lb.) for the conditions assumed, of between 5 and 6 per cent. It should be borne in mind that the 1,200-lb. equipment is special and therefore expensive. Very few stations have been designed for this pressure, and there is not nearly so great familiarity with the points involved in the design as there is with those of a 400-lb. station. It is reasonable to expect that the greater use of equipment for this pressure will result in cheapening its cost of

manufacture and design, while the 400-lb. design is practically standardized, and not much reduction can be expected in cost of either equipment or plant. From these considerations it would appear that the differential cost against high pressure will, if anything, tend to decrease in the future."

"Our expectations as to the improvement in economy at 1,200 lb. have been realized and, furthermore, we have found that the additional expense of the high-pressure equipment is justified by its greater power generating capacity. If the design is correct, a high-pressure station need not cost any more per unit of generating capacity than one designed for lower pressures. Mistakes in the design of high-pressure stations are very costly."

"The additional charges for high-pressure equipment are in part offset by the fact that the steam consumption per kilowatt-hour is less, thereby calling for a somewhat smaller condenser and somewhat less boiler-heating surface. In any event the increased cost of the 600-lb. plant (over the 400-lb.) amounts to only a few dollars per kilowatt."

"The ———— (1,300-lb.) high-pressure installation has a capital cost of 6.3 per cent over equivalent low-pressure capacity, and is 15 per cent more economical. Its ease of operation has been demonstrated many times, and future extensions will include 1,300 lb. pressure."

The cost of high-pressure industrial plants involves somewhat different considerations from those of the central station plant. These considerations, together with reliable cost figures, are contained in the following quotations of an eminent industrial engineer.

"The statement has been made many times that a 1,200-lb. plant costs no more than a 400-lb. plant, and no doubt this is true for central-station conditions where the reduction in water rate effected by high steam pressure affects the cost of condensers, condenser auxiliaries and circulating water supply. It is certainly not true for an industrial plant requiring both steam and power.

"A 400-lb. plant of 5,000 to 15,000 kw. capacity, in which the prime movers exhaust at 100 lb., can be built for $140 to $150 a kilowatt. A plant of the same size built for 1,200 lb. throttle pressure and 100 lb. exhaust pressure may cost $200 a kilowatt, particularly if it is necessary to provide evaporators for any considerable part of the feed-water supply. Experience has indicated that treated water may be considered at 400 lb. pressure, but becomes less and less practicable as the pressure is increased.

"Cost of power" may be as much a function of bookkeeping methods as it is of engineering design, but from construction and operating data now available, the following elements of cost may be definitely stated:

1. The capital cost should not exceed $200 per kw. installed. Of this $200 at least $75 would be required for a low-pressure plant to generate the same quantity of steam for process work, leaving the net investment chargeable to electric generation $125 per kilowatt.
2. The heat consumption should not exceed 4,500 B.t.u. per kw.-hr. net output.
3. Such a plant can be operated by five men per shift, of whom three would be required to operate a low-pressure steam plant, leaving two men chargeable to electric generation.

"Assuming a 15,000-kw. installation, 75,000,000 kw.-hr. annual output, 13,500 B.t.u. coal at $5 per ton, an average labor cost of $8 per day, and fixed charges and other overheads totaling 15 per cent, the cost of power would be about as follows:

Overhead, 15 per cent on $1,875,000	$281,250
Fuel, 75,000,000 kw.-hr. at ½ lb. per kw.-hr. at $5 per ton.	62,500
Labor, 6 men, 365 days at $8 per day	17,520
Maintenance	35,000
Miscellaneous expense	5,000
Total	$401,270
Cost per kw.-hr.	.00536

"The above represents power costs in a district of relatively high fuel costs, and generated in a plant of more expensive construction than is ordinarily considered for industrial plants. It is a plant, however, in which the pressure is high enough to require forged steel rather than riveted boiler drums, and high enough to preclude the use of treated water. Such a plant will avoid the most serious operating difficulties so far encountered in high-pressure plants."

The following quotations have a bearing on whether the assumption of base load conditions for a projected high-pressure plant is justifiable. The capacity factor (or annual plant factor) has an all important bearing on the economics of the situation.

"Consider the initial unit of a new power house built to meet the growing needs of a system containing certain other power houses. Naturally, this new unit, being more economical in coal consumption than the units in the older power houses, will be given the cream of the load. But the very giving of the cream of the load to the new unit affects adversely the load factor and output of all the original power houses, with the result that the total operating cost per kilowatt-hour of these power houses is increased. It is apparent, therefore, from a financial standpoint that consideration only of the fixed charges on the new installation and its operating cost under prescribed conditions cannot afford a complete solution of the problem. The effect of the new unit on the earning capacity of all the units previously installed should also be considered."

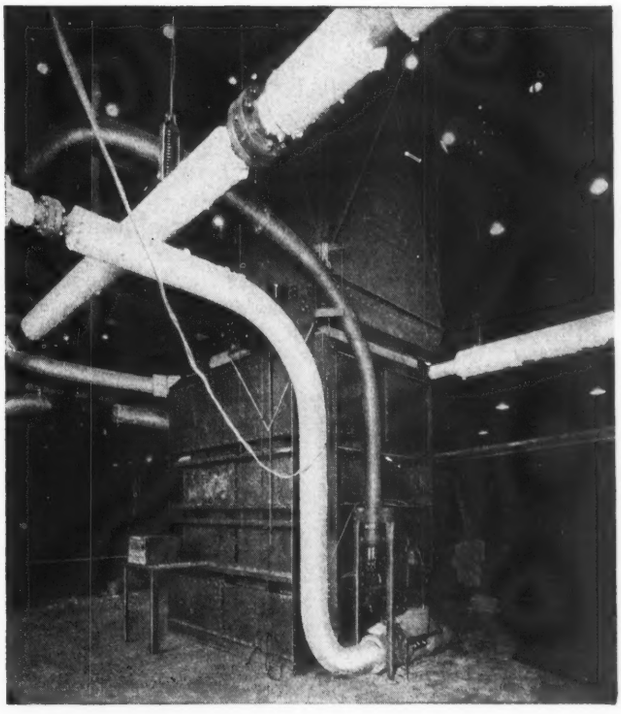
"The present tendency throughout the country toward complete interconnection of the power houses of a single system, and still further between systems of various companies, will, we believe, have a decided tendency to increase the capacity factor, making it less and less necessary to install spare units as plants are built, and in this way materially reducing the investment required, and increasing the active use of machinery installed."

"It appears, from the foregoing study of cycle efficiencies, that we are rapidly approaching a point where it is possible that the additional investment cost may not be warranted in obtaining high operating efficiencies. Until the present time it has been possible to obtain greater economy with so small an increase in investment cost that systems were warranted in building new stations for base load operation, permitting the older stations to operate over the peak load periods. As the limit of economy is approached, however, it appears that the differential between new and old station economy may become of so small a value that it would not pay to drop load from the old stations in order to carry base load on the new. In this case the building of peak load stations with low investment cost might be warranted at intervals as an alternate to the present policy of base load installations."

1,000-Degree Turbine *for Delray No. 3*

This experimental installation by the Detroit Edison Company marks the first attempt in this country to utilize high temperatures to improve turbine efficiency

By J. W. PARKER
Chief Engineer
The Detroit Edison Company



View of experimental superheater installed at the Trenton Channel Plant

THE Detroit Edison Company has recently placed with the British Thomson-Houston Company, Ltd., of Rugby, England, a contract for a 10,000-kw. two-cylinder, single-shaft steam-turbine generator to operate with an admission steam temperature of 1,000 deg. F. The inlet pressure will be 365 lb. per sq.in., conforming to the boiler pressure of the Delray Power House No. 3, in which the turbine is to be installed, while the exhaust pressure of one inch mercury absolute is that usually afforded in plants to which excellent circulating water is available.

While the Detroit Edison Company would have preferred postponing announcement of the plan of experimentation until more tangible accomplishment could be reported, it nevertheless seems advisable to make an authoritative statement now in order to correct misleading rumors that have become current.

The choice of the unusually high initial steam temperature, 1,000 deg., results from a study of the efforts of recent years to improve steam-turbine efficiency. Improvement of the steam cycle has been affected by two limitations. One is inherent, the fact that moisture in the turbine exhaust apparently must not exceed about 10 per cent, if rapid wear of blading and other parts in the lower pressure stages is to be avoided. Experience has, in fact, shown that entrained moisture may wisely be kept even below this amount. The other limitation has been the acceptance of 725 to 750 deg. as the highest practicable initial temperature, because of a cautious realization of the effects of temperatures higher than this on the strength of the metals ordinarily employed in superheaters, piping and turbines.

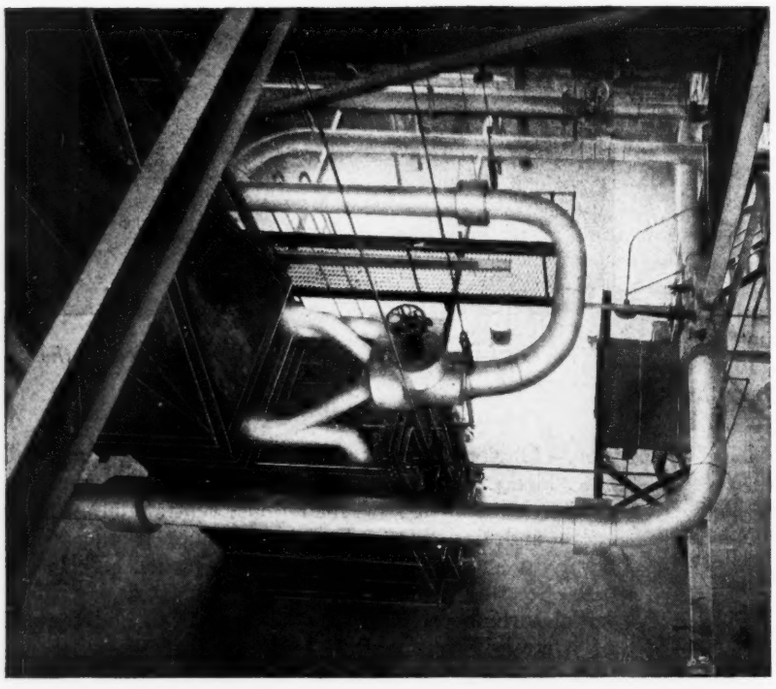
With such premises increase of steam pressure as a means of raising the ideal efficiency of the steam turbine reaches an upper limit at about 450 lb. unless resort is had to reheating at some intermediate point of expansion. Obvious disadvantages of the reheating method, the added difficulty of plant operation and the markedly greater investment have made most central-station managers and designers reluctant to adopt it except for highly developed high-load-factor plants. This has often determined the choice, in recent plant design, of an initial turbine pressure of 400 to 450 lb., and the use of a straight regenerative cycle without reheating, involving acceptance of a turbine efficiency inferior to that of the higher-pressure reheating plants.

The purpose of an experiment with higher-temperature steam, therefore, is simply to remove (as far as permitted by our present knowledge of the behavior of alloy steels at such temperatures) one of the conditions preventing the use of pressures higher than 450 lb. in the simple regenerative cycle.

The choice of an English turbine manufacturer was determined by a number of considerations. One of these was the thought that the project is essentially experimental, and if the turbine were to be designed and built in this country, it could not fail of diverting interest, both in our own minds and those of the manufacturers' engineers, from our established practice, which we believe should continue its tendency toward a standard design of turbine with major elements interchangeable within a large group of machines. Our long-time acquaintance with the work of F. Samuelson and the turbine building experience of the British Thomson-Houston Company, Ltd., of which he is engineer and manager, led us finally to the choice of that company. Nor were we unmindful of the experience English metallurgists have had with

the special steels that will be required to withstand the high temperature and of the proved skill of the British steel manufacturers.

It is planned to install the high-temperature turbine and its separately fired superheaters in a corner of the boiler room in the new Delray Power House No. 3, where space has been left available by the omission of one boiler from the first battery in the plant. Steam taken from the main superheated steam line at 400 lb. pressure and 725 deg. will be passed through this oil-fired superheater designed to deliver a maximum of approximately 83,000 lb. of steam per hour, and thence directly to the turbine. The machine itself will be placed close to the superheater in what would have been the longitudinal aisle between the first row of boilers and the auxiliary turbine room. So located, the run of pipe carrying 1,000 deg. steam will be short and condensing water connections will be simple. The energy generated will be delivered at 4,800 volts to the local alternating-current distribution system serving this district of the city.



Top of superheater showing provision made for expansion

As a preliminary step, an oil-fired superheater furnished by the Babcock & Wilcox Company, of New York, has been set up in a vacant boiler position at the Trenton Channel plant. This superheater is now in operation, taking 6,000 lb. of steam per hour from the main superheated steam lines at 700 deg. and delivering it to a high-temperature line at 1,000 deg. The delivery temperature will later be raised to 1,100 deg. After discharge from the superheater the steam is desuperheated to approximately 700 deg. by admixture with saturated steam of the same pressure and delivered to one of the plant auxiliary turbines whose operation affords a sufficiently continuous demand for steam to permit long experience runs. With this first installation it is intended to obtain preliminary experience of the effect of high superheat on tubes, pipes and pipe fittings, particularly with respect to valves and pipe joints. Data obtained on the rate of heat transfer through tube surfaces under high-superheat conditions, will also be of value in designing the more permanent equipment for Delray.

The photograph in the heading was taken during construction of the trial superheater at Trenton Channel. The piping with bends in the vertical plane is carrying steam to the inlet fitting on which the safety valve is mounted, while the piping with the 90-deg. bend in the horizontal plane is the outlet steam line carrying 1,000-deg. steam. Note the unusually heavy covering on this line. The other illustration is a view looking down upon the superheater. What appears to be wishbone-shaped pipe bends on top of the furnace are, in fact, extensions of the superheater tubes proper divided into two groups of two tubes each and given this peculiar shape to permit vertical expansion of the furnace tubes.

The entire surface receives its heat by radiation and consists of rows of vertical tubes set in the rear and side walls of the furnace. They are completely shielded on the fire side with refractory tile, and on the back are thickly insulated from the steel shell inclosing the whole structure. There are no tube headers, as in the usual type of superheater, but, instead, four tubes leading from the steam-inlet manifold carry the steam in four parallel paths through the superheater to the outlet fitting to which the tubes are welded. Each of these parallel paths consists of tubing butt-welded together to form one length of 275 ft. and arranged with suitable return bends at top and bottom of the furnace wall so as to form the grid of tubes in the back and sides of the furnace.

Whereas, the material used in the tubes near the inlet and forming the back wall of the furnace is plain low-carbon steel, the tube lengths in the side walls approaching the outlet are of special nickel-chromium steel, as are the discharge piping and flanges. This nickel-chromium steel can be welded satisfactorily and as far as can now be determined, is not brittle at high temperature and, therefore, is safe for a fiber stress of 1,000 lb. per sq.in. at 1,100 deg.

The pipe-line joints are at present Van Stone joints with which gaskets of several different materials are being tried. The flanges are so designed, however, that Sargol joints can be made if that appears necessary.

As it is too early even to predict what results will be obtained in this initial experiment, the foregoing statement is intended as a definition of the problem and an announcement of present plans rather than a description of equipment and materials to be used in the final experiment.

[*It is fitting that the Detroit Edison Company, which pioneered in the large boiler and was one of the two pioneers in the steam turbine, should take the lead in higher steam temperatures. Experience gained from the operation and design of this experimental installation will undoubtedly form the basis for wider application of high temperatures, especially among those plants whose designers have adhered to the range of moderate pressures.*—EDITOR.]

Events and Men in Power's Field

THE EDITORS WILL WELCOME THE CO-OPERATION OF READERS IN REPORTING INTERESTING HAPPENINGS, PERSONAL OR OTHERWISE

Many Celebrations Attendant on N.E.L.A. Covention

Dedication of Atlantic City Auditorium, Inauguration of Light's Golden Jubilee, Presence of Thomas Edison and Henry Ford, and Large Exhibition Are Features of 52nd Meeting

ACCOMPANIED this year by unusual celebrations the 52nd convention and exhibition of the National Electric Light Association will be held in Atlantic City, N. J., June 3-7. Dedication ceremonies of the Atlantic City Auditorium, where the convention and exhibition will be held, will precede the convention on May 31, while the initial celebration of Light's Golden Jubilee and the Diamond Jubilee of Atlantic city will take place during convention week. Thomas A. Edison, Henry Ford, Vice-President Curtis and many other prominent men are expected to participate in the festivities.

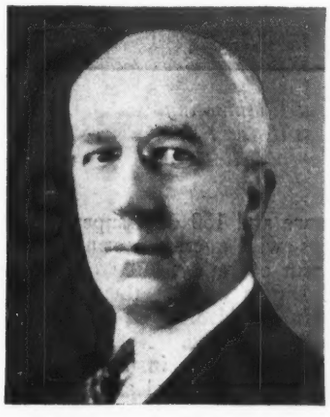
Monday, June 3, will be devoted to registration and inspection of the exhibition, followed by President Arkwright's reception and ball in the evening. The first general session will be held on Tuesday morning, at which W. J. Hagenah, vice-president of the Byllesby Engineering & Management Corporation, Chicago, Ill., will talk on regulation, and J. F. Owens, chairman of the Public Relations National Section, will speak.

Simultaneous sessions of the accounting and engineering sections will be held in the afternoon. At the engineering meeting, E. C. Stone, chairman of the Engineering National Section, will give a talk. "Shall We Grow Like Topsy or According to Plan" will be discussed by A. C. Marshall, vice-president of the Detroit Edison Company. F. R. Phillips, vice-president of the Duquesne Light Company, of Pittsburgh, will talk on the topic, "Engineers and Engineers," and a paper on "The Development of Electrical Standards" will be presented by L. L. Elden, of The Edison Electric Illuminating Company of Boston.

At the session on Wednesday morning the electrification of railroads will be discussed by Britton I. Budd and George D. Gibbs. M. S. Sloan, president of the New York Edison Company, will talk, and J. J. Carty, vice-president of the American Telephone & Telegraph Company, will present a paper on "Science and Progress in the Industries." The Commercial National Section, with M. E. Sampsell, chairman, will hold its session in the afternoon, at which reports of the lighting, power and merchandising bureaus will be heard.

Thomas A. Edison and Henry Ford will be introduced at the general session on Thursday morning, and John W. Lieb will preside. C. L. Edgar, president of The Edison Electric Illuminating Company of Boston, will speak. A meeting of the Public Relations National Section will be held in the afternoon with J. F. Owens, chairman, presiding.

An attractive program of entertainment is planned in connection with the Public Policy Committee meeting on Thursday evening. Albert Spalding, violinist, Frieda Hempel, soprano, and



P. S. Arkwright
President of the N.E.L.A.

B. A. Rolfe's orchestra will be heard. President Arkwright will present the Charles A. Coffin award, and Julius H. Barnes will talk on "America's Fifty Years of Aladdin." This program will be broadcast through the courtesy of the National Broadcasting Company.

"Rural Electrification," by J. R. Howard, past-president of the American Farm Bureau Federation and "Industrial Development in the United States," by J. L. Madden, vice-president of the Metropolitan Life Insurance Company, will be presented at the fourth general session on Friday. W. L. Ransom will discuss "The Public Right as to Electric Rates," and Bernard F. Weadock, special counsel before the Federal Trade Commission, will speak. After the reports of various committees and election of officers the convention will adjourn at about one o'clock.

Spot ⁎ News

ACCORDING TO *the latest published report of the American Uniform Boiler-Law Society, eighteen states, the District of Columbia, Hawaii and fifteen large cities in the United States have already adopted Section 1, relating to power boilers, of the A.S.M.E. Boiler Construction Code.*

* * *

COAL *as a source of power is yielding to oil and water, according to a review of the international coal trade situation issued by the Department of Commerce, in which statistics are quoted showing that although the world power output increased 20 per cent, only one-fifth of that increase was due to coal, and the remaining due to oil and water power.*

* * *

A DUPLICATE *of the 94,000-kw. unit installed last year is being added to the Long Beach steam plant of the Southern California Edison Company. When in service in 1930, the new unit will bring the capacity of the plant to well over a million horsepower.*

* * *

OWING TO CONSOLIDATION *the number of electric power plants serving the United States has decreased from 6,355 to 2,135 in the period of five years between 1922 and 1927.*

* * *

A COURSE *in Public Utility Management and Economics will be given this year at the summer session of Harvard Graduate School of Business Administration, Cambridge, Mass. The course runs from July 1 to August 10, and is open to those interested in utilities.*

* * *

A DIESEL-ENGINE *electric generating plant with an ultimate capacity of 7,500 hp. will be built by the Moaesto Irrigation District, of Modesto, Calif. Starting this year, units of 1,500 hp. will be installed each year until the plant is completed.*

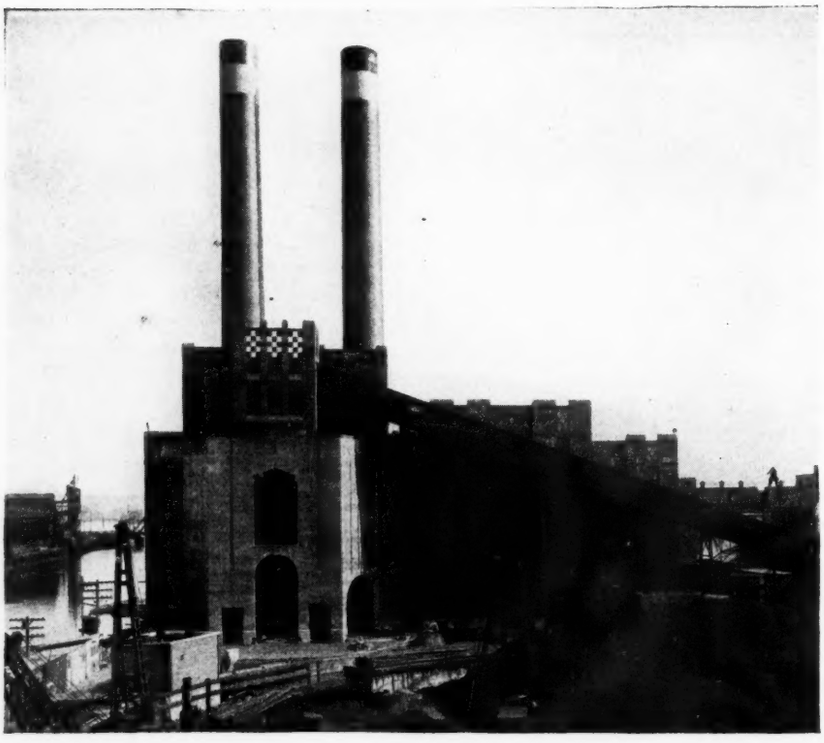
* * *

THE ELECTRIFICATION *of Poland is planned in a $100,000,000 construction program of steam generating plants and transmission lines to be financed by American interests.*

* * *

THE NEW ADDITION *to the main generating station of the San Diego Consolidated Gas & Electric Company, San Diego, Calif., was formally opened recently. The new building houses a 28,000-kw. turbine with auxiliaries.*

Engineers Inspect English Station at New Haven, Conn.



North elevation of the United Illuminating Company's new station as it nears completion

OVER one hundred engineers were the guests on May 17 of Westcott & Mapes, Inc., engineers and architects, at an inspection of the new English Station of the United Illuminating Company, of New Haven, Conn., the first unit of which recently went into operation. The visitors were mainly members of the New Haven Section of the A.S.M.E., the Connecticut Sections of the A.S.C.E. and A.I.E.E., and representatives of several equipment manufacturers.

The new station is named in honor of James English, who for 45 years has been directing head of the company. When completed, it will have a capacity of 75,000 kw. It is being built in two sections, each of which provides space for three 15,625-kva. generating units and six 15,112-sq.ft. cross-drum boilers. The first section, now nearing completion, contains two generating units and its full complement of boilers. Construction work on the second section is in progress.

The English Station is located on a made-ground site at the rear of the company's present Grand Avenue Station. With harbor frontage on three sides the site provides excellent facilities for supplying condensing water and receiving coal. Between the two plants there is adequate space for the storage of 13,000 tons of coal.

Steam will be generated at 375 lb. pressure and 150 deg. superheat. The boilers will operate normally at 200 per cent of rating, but stoker capacity sufficient to operate them at 300 per cent is provided. The stokers are 23 tuyeres underfeed. No economizers or air preheaters are installed. With the exception of the coal and ash-handling equipment and small auxiliary pumps, all the station auxiliaries are steam driven, and the heat balance is obtained through the operation of duplex steam and electrically driven exciters. The change from one drive to the other is made automatically in accordance with the demand for exhaust steam to maintain a feed water temperature of 220 degrees.

In designing the station simplicity and reliability of service have been given first place. The boilers and generating units have been selected to meet the low load factor (around 30 per cent) and provide flexibility and ease of operation.

A.S.R.E. to Hold Spring Meeting at Penn. State College

The 16th spring meeting of the American Society of Refrigerating Engineers will be held at Pennsylvania State College, State College, Pa., June 20-22. An interesting program of technical papers and entertainment is planned for the occasion. All the sessions will be held in Varsity Hall, while the exhibits of equipment will be displayed in the mechanical engineering laboratory and thermal plant of the college.

The first technical session on Thursday morning, June 20, will be devoted to ice-plant design, a feature of which will be a paper on "Oil Engines for Ice Plant Compressors." In the afternoon two papers entitled, "Climatic Studies; Effect on Refrigeration" and "Instruments for Engineering Measurement; Electrical Method," will be presented.

A joint session with the Railroad Division of the A.S.M.E. is scheduled for Friday to consider the problems of transit refrigeration. Among the papers to be given is "Machinery for Pre-Cooling Stations." Saturday's session will be devoted to household refrigeration. Demonstrations of heat transmission investigations and testing machinery together with the Refrigerating Engineering Exhibit add to the attractiveness of the technical program.

Convention of A. I. & S. E. E. at Pittsburgh, June 17-21

The 25th annual convention and exposition of the Association of Iron and Steel Electrical Engineers will be held at the William Penn Hotel in Pittsburgh, Pa., June 17-21. Many of the papers to be presented at the technical sessions and much of the Iron and Steel Exposition should prove interesting to engineers in the power field.

At the Electrical Engineering Division session on Monday, June 17, the Electric Welding Committee with S. S. Wales as chairman will give a report on "Electrical Welding." "The Incorporation of Safety into Electric Station Design," by V. R. Bacon will be presented at the afternoon session of the Safety Engineering Division.

On Tuesday morning the Special Committee on Bearings with F. D. Egan, chairman, will give a paper on "Anti-Friction Bearings for Mill Type Motors." This will be followed by "Electrical Developments in the Iron and Steel Industry," by the Electric Development Committee, Walter H. Burr, chairman. L. A. Umansky will present "The Selection of Electric Drives" on Wednesday morning.

The Combustion Engineering Division, on Tuesday, Wednesday and Thursday mornings, will present eight papers of particular interest, as follows: "Utilization of Steam," by L. C. Edgar; "Feed Water Treatment," by S. E. Hall; "Efficiency Versus Economy," by H. G. Ebdon; "The Utilization of Waste Gases," by Walter Flanagan; "Liquid Fuel," by R. C. Vroom; "Economizers," by Walter Keenan; "Gas Producers," by F. E. Leahy; and "The Application of Pulverized Fuel in Industrial Furnaces," by W. O. Renkin. A paper on "Time Current Control for D.-C. Motors," by D. C. Wright, closes the technical sessions on Friday.

OBITUARY

HENRY W. BLAKE, senior editor of the *Electric Railway Journal*, died May 20, at his home in Englewood, N. J. He was in his sixty-fourth year.

Mr. Blake was born in New Haven, Conn., and was graduated from Yale in 1886. He then studied electrical engineering at the Massachusetts Institute of Technology and became active in electric railway and motor construction.

In 1891 he joined the staff of *The Street Railway Journal*, a publication originally devoted to horsecar transportation, of which he became editor in 1894. The journal was later devoted to the electric railway transport trade and its name was changed to the *Electric Railway Journal* in 1908. Mr. Blake had been associated with it since that time.

He was a member of the American Institute of Electrical Engineers, the Engineers Club and the Englewood and Knickerbocker Country Clubs. He was an associate member of the Union Internationale de Tramways of Brussels.

PERSONALS

DAVID B. RUSHMORE, former consulting engineer of the General Electric Company, has been appointed by President Hoover to represent the United States at the World Power Conference meeting in Barcelona, Spain. Influential in every line of industrial power, Mr. Rushmore has been a delegate to previous World Power Conference meetings.

NEIL CURRIE, JR., managing engineer of the motor department of the Pittsfield Works of the General Electric Company for the last five years, has been named manager of the Philadelphia Works.

JACK MALONE, Utah Power & Light Company, Salt Lake City, has resigned to join the staff of the Consumers Central Heating Company, Tacoma, Wash.

JOHN J. O'BRIEN has been elected president of the Philadelphia Company to succeed A. W. Robertson who resigned some time ago to become chairman of the board of the Westinghouse Electric & Manufacturing Company at Pittsburgh.

BUSINESS NOTES

THE AIR PREHEATER CORPORATION, of New York City, has recently opened an office in the Tacoma Building, Tacoma, Wash.

THE JEFFREY MANUFACTURING COMPANY, of Columbus, Ohio, announces the incorporation of the Jeffrey Manufacturing Company, Ltd., at Montreal, Quebec. Besides the main office and works in Montreal, this new Canadian company will have a branch office in Toronto and a warehouse in Calgary, Alberta.

THE READING STEEL CASTING COMPANY recently moved its New York office to new quarters in the New York Central Building, 230 Park Avenue. The New York City warehouse of the same company is now at 333 Hudson Street.

THE HAHN ENGINEERING COMPANY, now a division of the Lancaster Iron Works, Inc., Lancaster, Pa., has opened offices in New York at 501 Fifth Avenue in charge of C. H. Lank, and in Philadelphia, in the Land Title Building, in charge of G. E. Kille.

Coming Conventions

American Boiler Manufacturers Association. Annual meeting, June 3-5, Sky Top Lodge, Cresco, Pa. A. C. Baker, secretary, 801 Rockefeller Blvd., Cleveland, O.

American Institute of Electrical Engineers, annual summer convention, Swampscott, Mass., June 24-28. Pacific Coast Regional Meeting, Santa Monica, Calif. Sept. 3-6. F. L. Hutchinson, secretary, 33 West 39th St., New York City.

American Society of Heating and Ventilating Engineers, annual summer meeting, Bigwin Inn, Lake of Bays, Ontario, Can., June 26-28. A. V. Hutchinson, secretary, 29 W. 39th St., New York City.

American Society of Mechanical Engineers, spring meeting, Salt Lake City, Utah, July 1-4, 1929. Calvin Rice, secretary, 29 West 39th St., New York City.

American Society of Refrigerating Engineers. Summer convention at Penn State College, State College, Pa., June 20-22. Secretary, W. H. Ross, 35 Warren St., New York City.

American Society for Testing Materials. Annual meeting, June 24-28, at Chalfonte-Haddon Hall, Atlantic City, N. J. Secretary, C. L. Warwick, 1315 Spruce St., Philadelphia, Pa.

Association of Iron and Steel Electrical Engineers, 25th annual convention, Pittsburgh, Pa., June 17-22, 1929. A. A. Stewart, secretary, Pittsburgh Steel Co., Monnessen, Pa.

Canadian Electrical Association, annual convention, to be held June 19-21, at St. Andrews-by-the-Sea, N. B. H. M. Lyster, secretary, Power Building, Montreal, Que.

National Association of Power Engineers, 47th annual convention, Chattanooga, Tenn., Sept. 9-13. Fred Raven, secretary, 417 South Dearborn St., Chicago, Ill. N. J. State Association annual meeting, June 1-2, at Eagles Hall, Elizabeth, N. J. Conn. State Association annual meeting, June 14-15, at the Stratfield Hotel, Bridgeport, Conn. New England States Association annual meeting, June 20-21, at Fitchburg, Mass.

National Board of Boiler and Pressure Vessel Inspectors. Annual meeting, June 18-20, at Detroit, Mich. Secretary, C. O. Myers, 14 Commercial Nat'l Bank Bldg., Columbus, Ohio.

National Electric Light Association. Annual meeting at Atlantic City, N. J., June 3-7. A. J. Marshall, secretary, 420 Lexington Ave., New York City.

National Exposition of Power and Mechanical Engineering. Grand Central Palace, New York City, Dec. 2-7. Charles F. Roth, manager, Grand Central Palace, New York City.

National District Heating Association. Annual meeting, June 11-14, at the Hotel Statler, Detroit, Mich. Secretary, D. L. Gaskill, Greenville, Ohio.

National Oil and Gas Power Meeting. Under auspices of A.S.M.E. Oil and Gas Power Division and the Pennsylvania State College. June 24-27, State College, Pa. F. G. Hechler, chairman, State College, Pa.

Northwest Electric Light and Power Association. 22nd Annual Convention, June 26-29, 1929, at Seattle, Wash. Registration and information: J. J. Hayes, Westinghouse Electric & Manufacturing Company, Seattle, Wash.

Universal Craftsmen Council of Engineers. Annual meeting in Detroit, Mich., Aug. 6-10, at the Book-Cadillac Hotel. Grand Secretary, T. H. Jones, 33 Linden Ave., Cherrydale, Va.

World Engineering Conference, Tokyo, Japan, October, 1929. All correspondence should be addressed to the Secretary, World Engineering Conference, Nikon, Kogyo Club, Morunouchi, Tokyo.

FUEL PRICES

COAL

The following table shows the trend of the spot steam market in various coals, f.o.b. mines; mine run, except Pittsburgh gas slack:

Bituminous (Net Tons)	Market Quoting	Price per Ton
Navy Standard	New York	$2.35 @ $2.40
Kanawha	Columbus	1.25 @ 1.45
Smokeless	Cincinnati	1.75 @ 2.00
Smokeless	Chicago	1.75 @ 2.25
S. E. Kentucky	Chicago	1.35 @ 1.60
Steam	Pittsburgh	1.55 @ 1.65
Gas Slack	Pittsburgh	1.00 @ 1.15
Big Seam	Birmingham	1.25 @ 1.50
Anthracite (Gross Tons)		
Buckwheat	New York	$2.75
Barley	New York	1.50

FUEL OIL

New York—May 23, f.o.b. Bayonne, N. J., 28@34 deg., Baumé, industrial use, tank-car lots, 5½c. per gal.; 36@40 deg., furnace, tank-car lots, 6c. per gal.

St. Louis—May 16, tank-car lots, f.o.b. St. Louis, 24@26 deg., $1.3959 per bbl. or 42 gal.; 26@28 deg., $1.4459 per bbl.; 28@30 deg., $1.4959 per bbl.; 30@32 deg., $1.5459 per bbl.; 32@36 deg., gas oil, 4.276c. per gal.; 38@40 deg., distillate, 5.23c.

Pittsburgh—May 13, f.o.b. local refinery, 30@34 deg., fuel oil, 5c. per gal.; 36@40 deg., 5.5c. per gal.

Philadelphia—May 16, 13@19 deg., $1.05@$1.10 per bbl. or 42 gal.; 22 deg. plus, $1.627@$1.677 per bbl.; 23@27 deg., $2.10@$2.15 per bbl.

Cincinnati—May 21, tank-car lots, f.o.b. local refinery, 24@26 deg. Baumé, 5c. per gal.; 26@30 deg., 5.25c. per gal.; 30@32 deg., 5.5c. per gal.

Chicago—May 18, tank-car lots, f.o.b. Oklahoma, freight to Chicago, 90c. per bbl. or 42 gal.; 22@26 deg., 60c. per bbl.; 26@30 deg., 80c. per bbl.; 30@32 deg., $1.00 per bbl.

Boston—May 20, tank-car lots, f.o.b. 12@14 deg. Baumé, 4.35c. per gal.; 28@32 deg., 5.5c. per gal.

Dallas—May 18, f.o.b. local refinery, 26@30 deg., $1.35 per bbl. or 42 gallons.

New Plant Construction

COMPILED BY THE McGRAW-HILL BUSINESS NEWS DEPARTMENT, WHICH IS PREPARED TO FURNISH A MORE COMPLETE DAILY SERVICE TO THOSE WHO WISH IT

Calif., San Francisco—Constructing Quartermaster U. S. Government, is having plans prepared for the construction of a power plant, 400 hp. boiler plant, etc. at Fort McDowell.

Calif., Santa Barbara — Cottage Hospital, awarded contract for a 4 story addition to hospital including power and heating plant, etc. to A. MacKeller, 112 East Sola St. Estimated cost $800,000.

Colo., Denver—Liquid Carbonic Co., 3100 Kedzie Ave., Chicago, Ill., awarded contract for the construction of a dry ice plant here to Austin Co., 16112 Euclid Ave., Cleveland, O. Estimated cost $150,000.

D. C., Washington—Treasury Dept., Office of J. A. Wetmore, Acting Supervising Archt., will receive bids until June 7 for the construction of a pumping plant for U. S. Treasury Building.

Ga., Atlanta — Georgia Power Co., Electric a steam driven electric generating plant on Chattanooche River near here to Empire Construction Co., Electric Bldg. H. F. Armstrong, in charge. Estimated cost $10,000,000. W. P. Hammond, c/o Alabama Power Co., Birmingham, Ala., is engineer.

Ill., Chicago—Commonwealth Edison Co., 72 West Adams St., awarded contract for addition to frequency changer house at Fisk and 22nd Sts. to Otto Randolph Inc., 53 West Jackson Blvd. Estimated cost $300,000.

Ill., Rantoul—U. S. Army, Chanute Field, is having preliminary plans prepared for improvements to army air field including hangar, field shop, central heating plant, etc. Estimated cost $252,700. Private plans.

Ill., Scott Field (mail Belleville) — U. S. War Dept., is having preliminary plans prepared for improvements to airfield to include central heating plant, shops, etc. here. Estimated cost $508,000. Private plans.

Ill., Tilton—Standard Oil Co. of Indiana, Whiting, Ind., awarded contract for the construction of boiler houses, warehouses, etc. here to Clarke Construction Co., Denville, Ill. Estimated cost $135,000.

Ind., Michigan City—Holabird & Root, 333 North Michigan Ave., Archts., will receive bids until June 3 for the construction of an electric generating station to include turbine, boiler and pump rooms, etc. here, for Northern Indian Public Service Co., Hammond. Sargent & Lundy, 72 West Adams St., Chicago, are engineers.

Ia., Ottumwa—Iowa Southern Utilities Co., Centerville, awarded contract for addition to power plant to Federal Engineering Co. Centerville. Estimated cost $250,000.

Ia., Shanandoah—Iowa—Nebraska Light & Power Co., plans extensions and improvements to light plant. Estimated cost $150,000.

La., Alexandria—Refrigerator Products, Inc., c/o E. Harding, Pres., 10th and Murray Sts., awarded contract for the construction of an 85 x 102 ft. ice cream and cold storage plant to J. V. Pellerin. Estimated cost $85,000.

Md., Laurel—D. C. Commissioners, Washington, D. C., will soon award contract for remodeling power plant at District Training School here.

Mass., Gay Head—U. S. Lighthouse Division, plans the construction of an electric power plant. Private plans.

Mass., Lynn—Lynn Gas & Electric Co., 90 Exchange St., awarded contract for the construction of a turbine room and electric bay, etc. on Broad St. to P. J. McNaughton Inc., 755 Boylston St., Boston.

Mich., Grosse Pointe Park—Dept. of Public Works, Detroit, will soon award contract for the construction of a sewage pumping station, etc. Estimated cost $25,000. F. A. Fellows, City Hall, Detroit, is engineer.

Minn., Wayzata — City, G. L. Mullens, awarded contract for the construction of a waterworks system including pumping station, etc. to Phelps-Drake Co., 623 Metropolitan Life Bldg., Minneapolis. Estimated cost $43,000.

Mo., Lee Summit—W. B. Rollins & Co., 339 Railway Exchange Bldg., Kansas City, made preliminary surveys for the construction of municipal electric light plant. Maturity indefinite.

Mo., St. Louis—Laclede Gas Light Co., G. B. Evans, 1001 Laclede Bldg., 11th and Olive Sts., plans the construction of a 45 x 70 ft. pumping station. Management & Engineering Corp., 327 South La Salle St., Chicago, Ill., is engineer, also awarded contract for a gas holder, 10,000,000 cu.ft. capacity to Bartlett-Haywood Co., 200 Scott St., Baltimore, Md. Total estimated cost $500,000.

Neb., Broken Bow—City, W. D. Gardner, Clk, awarded contract for the construction of a 51 x 51 ft. power house including generating equipment, electric switchboard, etc. to Fairbanks, Morse & Co., 9th and Harney Sts., Omaha.

N. J., Hillsdale—Dept. of Institutes & Agencies, State Office Bldg., Trenton, is having preliminary plans prepared for the construction of a 2 story dormitory and power plant. Estimated cost $150,000. Guilbert & Betelle, 20 Branford Pl., Newark, are architects.

N. J., Jersey City—Brunswick Laundry, 222 Tonnele Ave., will receive bids about June 3 for the construction of a power house and boiler room at 68-70-72 Liberty Ave. Estimated cost $150,000. P. Simeone & Co. Inc., 22 Journal Sq., are architects and engineers.

N. Y., New York—New York Edison Co., 4 Irving Pl., plans extension to steam power plant to house equipment for desuperheating steam at 686 First Ave. Estimated cost $80,000. Private plans.

N. Y., New York—Manhattan Refrigerating Co., 525 West St., will not construct a refrigeration plant. Gunvald Aus Co., 244 Madison Ave., is engineer. Project abandoned.

Okla., Oklahoma City — Oklahoma-Biltmore Hotel Co. awarded contract for a 25 story hotel on Grand St., to Reinhart & Donovan, Commerce Exchange Bldg. Estimated cost $3,000,000. Steam heating and refrigeration systems, boilers, pumps, elevators, etc. will be installed.

Tex., San Angelo—West Texas Utilities Co., plans the construction of a power dam, 6,000 ft. long and 45 ft. high on the Concho River, six mi. south of here. Estimated cost $500,000. Private plans. Work will be done by owners' forces.

Va., Lynchburg—Randolph Macon Woman's College, awarded contract for the construction of a music building and power house to John P. Pettyjohn & Co., 212 8th St. Estimated cost $210,000.

Wis., Prairie Du Chien—Prairie De Chien Woolen Mills, awarded contract for the construction of a power plant and boiler house to G. J. Lingst, Prairie Du Chien.

B. C., Ruskin — British Columbia Electric Railway Co., 425 Carroll St., Vancouver, awarded contract for hydraulic turbine to Dominion Engineering Works, Ltd.; generator to Canadian Westinghouse Co., Hamilton, Ont.; four step-up transformers to Canadian General Electric Co., Toronto, in connection with generating unit of Ruskin hydro-electric plant here. Estimated cost $690,000.

Ont., Chapleau — Chapleau Electric Light & Power Co., awarded contract for the construction of a hydro-electric development including power house, penstocks, two 275 hp. units, etc. to Dominion Construction Co., Reford Bldg., Toronto. Estimated cost $100,000.

Ont., Saulte Ste. Marie—Abitibi Power and Paper Co., Ltd., plans extensions and improvements to plant. Estimated cost $1,000,000.

Equipment Wanted

Engine—City of Russell, Kan., $35,000 bond election May 28, for engine for electrical system.

Engine, Generator, Etc.—Treasury Dept., Office of J. A. Wetmore, Supervising Archt., Washington, D. C., will receive bids until June 12 for engine, generator, etc. at U. S. M. H. Fort Stanton, N. M.

Pump — Pinehurst Community, Pinehurst, Wash., plans to purchase automatic pump, etc. for proposed waterworks improvements. Estimated cost $20,000.

Pumping Equipment—Bd. of Council, Boro Hall, Oakland, N. J., will soon receive bids for pumping equipment, etc. for proposed waterworks.

Pumping Unit—City of Wheeler, Tex., pumping unit, etc. for proposed waterworks improvements. Estimated cost $45,000.

Pumps, Etc.—City of Watsonville, Calif. two 1600 g.p.m. deep well pumps for Fort Street auxiliary plant.

Pumps, Etc.—City of Vaiden, Miss., J. C. Calhoun, Clk., will receive bids until June 3 for two motor driven pumps, etc. for proposed waterworks.

Pumps, Etc.—City of Cedar Rapids, Ia., L. J. Storey, Clk., will receive bids until June 12 for pumps, etc. for proposed waterworks improvements.

Pumps, Etc.—City of Woodriver, Ill., plans to purchase pumps, etc. for proposed waterworks improvements. Estimated cost $40,000 to $72,000.

Refrigeration Equipment—Ontario Dairies Ltd., L. S. Beninger, Mgr., Wingham, Ont., refrigeration and pasteurizing equipment, etc. for proposed creameries.

Industrial Projects

Calif., Los Angeles — CREAMERY AND BOTTLING PLANT—Safeway Stores Inc., 1925 East Vernon St., awarded contract for a 3 story, 100 x 130 ft. creamery and bottling plant at Alameda and Vermont Sts. to W. Mellema, Beau Arts Bld. Estimated cost $200,000.

D. C., Washington—CEMENT PLANT—International Cement Corp., 342 Madison Ave., New York, awarded general contract for the construction of a cement plant 60,000 bbl. capacity with complete packing plant, 8 silos, each 80 ft. high, elevating and conveying equipment, etc. at 34th and K Sts. here to Rust Engineering Co., District National Bank Bldg., Pittsburgh, Pa. Estimated cost $300,000 to $400,000.

Ill., Springfield—FLOUR MILL—Pillsbury Flour Mill Co., 302 Metropolitan Life Bldg., Minneapolis, Minn., awarded contract for an 8 story 65 x 351 ft. mill including 80 x 200 ft. cereal mill, 80 x 240 ft. warehouse, 56 grain storage tanks, 1,300,000 bu. capacity here to Jones-Hettelsater Construction Co., 600 Mutual Bldg., Kansas City, Mo. Estimated cost $1,000,000

Ind., Richmond — GEAR FACTORY — Automotive Gear Works, awarded contract for a 2 story, 60 x 124 ft. gear factory to C. M. Guepel Construction Co., 1017 Hume-Mansion, Indianapolis. Estimated cost $65,000.

Mass., Chelsea—PIPE FACTORY—Inter-Continental Pipe & Mining Co., H. Kennedy, Pres., 475 5th Ave., New York, N. Y., plans the construction of a plant here. Estimated cost $3,000,000. Private plans.

Mich., Marysville — METAL WORKING PLANT — Pressed Metals of America, Inc., awarded contract for a 2 story, 50 x 70 ft. addition to plant for the manufacture of pressed metal products for automobiles on Gratiot Rd. to Schott Bros. & Weber, Mt. Clemens. Estimated cost $50,000.

Mo., North Kansas City (br. Kansas City)—HARVESTER COMBINE FACTORY — Curtis Harvesters, Inc., plans the construction of a 2 story, 225 x 400 ft. factory here. Estimated cost $250,000. McKecknie & Trask, Bd. of Trade Bldg., are architects.

Mo., St. Louis—DISPLAY FIXTURE FACTORY—Multiplex Display Fixture Co., 918 North 10th St., will soon award contract for a 2 story, 57 x 105 ft. factory. C. G. Schoelch & Co., 1247 Syndicate Trust Bldg., are architects.

N. J., Newark—STEEL BARREL PLANT—Maurer Steel Barrel Co., Long Island, City, N. Y., awarded contract for a 1 story, 175 x 300 ft. factory at Ave. L and St. Charles St. here to Austin Co., 120 Broadway, New York, N. Y. Estimated cost $150,000.

N. J., Passaic—PLANT and BOILER HOUSE—Campbell Morrell & Co., 1 Park Pl., will build a 3 story, 58 x 110 ft. plant and 90 x 40 boiler house on First St. Estimated cost $65,000. Private plans. Work will be done by separate contracts.

O., Cleveland—AIRPLANE FACTORY—Great Lakes Aircraft Corp., C. F. Van Sicklen, Sales Mgr., 16800 St. Clair Ave., awarded contract for addition to factory to Austin Co., 16112 Euclid Ave. Estimated cost $75,000.

S. C., Rock Hill — FACTORY and BOILER HOUSE—Rock Mill Printing & Finishing Co., will soon award contract for a 2 story, 225 x 478 ft. factory and 52 x 75 ft. boiler house.

Ont., Copperclife—COPPER RERFINERY—International Nickel Co., awarded contract for the construction of copper refinery to Fraser, Brace Ltd., Craig St., Montreal, Que. Estimated cost $4,000,000.

Ont., Harrow — CANNING PLANT and BOILER HOUSE — W. Clarke & Co., awarded contract for a 3 story addition to canning plant including new boiler house, boilers, 300 ton silo for coal storage, etc. to Wilson & Muxlow, c/o owner. Estimated cost $100,000.

Ont., Windsor — STEEL PLANT — Swedish Crucible Steel Co., 1165 McDougall Ave., awarded general contract for the construction of a 1 story steel plant on Hanna Ave. to Allan & Earl, Equity Chambers. Estimated cost $100,000.

B. C. Vancouver (AIRPLANE FACTORY)—Boeing Aircraft Ltd. of Canada, Georgia St. W., awarded contract for a 1 story airplane factory to Austin Co. of Canada, 470 Granville St. Estimated cost $100,000.